Chapter 2

HVAC Control System Equipment, Equipment Uses and HVAC Control Loops

2-1. General

a. The design of HVAC control systems is implemented by defining the operating modes of the HVAC equipment, defining the control loops required, and selecting the control system equipment to be used in the loop. The process of selecting the control system equipment includes calculations by the designer to specify the flow capacity of control devices, the physical size of control devices and the electric service required.

b. This chapter describes the operating modes, process variables, control, modes, control-system devices and their features, control system equipment applications, and interconnection of control devices. This chapter provides criteria and guidance for selecting and sizing control devices.

2-2. Control of HVAC-system operating modes and process variables.

a. Control-system operating modes. Control systems start and stop the HVAC systems according to a time schedule, and at specific outside-air temperatures and specific indoor temperatures. In addition, the control systems operate the HVAC systems in the following modes of operation:

(1) Occupied mode is initiated automatically to allow HVAC systems to start in sufficient time to bring the space to the proper temperatures at the start of occupancy.

(2) Ventilation-delay mode is initiated automatically to prevent the use of outside air when the unit is started prior to occupancy, to cool down or warm up the area served.

(3) Unoccupied mode is initiated automatically to prevent unnecessary operation of HVAC-system equipment during periods of non-occupancy except for special purposes such as operation to maintain minimum space temperatures for freeze protection.

(4) Heating or cooling modes are initiated manually to provide either heating or cooling media to HVAC equipment.

b. Control of HVAC-system process variables. While the HVAC systems are in operation, the process variables commonly sensed and controlled by HVAC control systems are:

(1) Temperature.
(2) Relative humidity.
(3) Static pressure of air.
(4) Differential pressure of air.
(5) Air-flow rate.

c. Constraints on *process variables by operating modes.* The constraints placed on the control of HVAC process variables by the operating modes are:

(1) Cooling and humidification are shut off during the unoccupied mode.

(2) Outside air is not supplied to the space during the unoccupied and ventilation-delay modes.

d. Modulating control. The amount of heat delivered to a space (or removed from a space) from certain types of HVAC equipment is regulated by varying the heat-exchanger capacity from zero to one-hundred percent in response to the variation of a continuous, gradual input signal. This is called modulating control. Heat-exchanger control valves, mixing dampers, fan-inlet vanes, variable-speed drives, and humidifier valves are examples of HVAC equipment that are controlled by modulating control.

e. Two-position control. The amount of heat delivered to a space from certain types of HVAC equipment is controlled by turning the equipment on and by shutting the equipment off. This type of control is also called on-off control. Examples of 2-position control are the starting and stopping of the fans of unit heaters and fan-coil units by room thermostats to maintain space temperature, and the opening and closing of shut-off dampers when fans are started and stopped.

2-3. Control-system equipment.

a. Control valves.

(1) Control valves are used to regulate the flow of fluids in piping systems by compressing and releasing a valve spring to move a valve-closure disk or plug toward or away from the closure seat of a flow port. The valves are used both in modulating and in 2-position control applications.

(2) Examples of the use of modulating control valves are:

(a) Heating- and cooling-coil control valves.
(b) Converter steam-control valves.
(c) Humidifier control valves.
(d) Perimeter-radiation-systems zone valves.

(3) Examples of the use of 2-position control valves are:

(a) Dual-temperature water-system changeover valves.
(b) Shut-off valves used in fan-coil unit coils.

(4) Control valves are classified according to their flow-regulating body patterns. A 2-way valve restricts fluid flow in one direction, because it has one inlet and one outlet; a 3-way valve restricts flow in two directions. The designer will use 2-way control valves for controlling the following types of HVAC equipment:

(a) Convertors.
(b) Radiators.
(c) Coils served by variable-volume pumping systems.
(d) Steam coils.

(5) A 3-way mixing valve has two inlets and one outlet, and a 3-way bypass (diverting) valve has one inlet and two outlets. For the systems shown in this manual, the 3-way mixing valve is used in both flow-mixing and flow-diverting type applications, except that a 3-way bypass valve is used as a dual-temperature changeover control valve.

(6) In the flow-mixing application, the 3-way valve is used to mix heated primary flow, from a boiler or a converter, with system return flow to produce system secondary supply, for the purpose of controlling temperature. When used on the return line from a coil, one of the 3-way valve's inlets is from the coil, and the other inlet is from the bypass around the coil. The designer may choose to use 3-way mixing valves in lieu of the 2-way valves shown in this manual for controlling the following types of HVAC equipment to prevent deadheading of pumps:

(a) Coils served by constant-volume pumping systems.

(b) As means of pump pressure relief in variable-volume pumping systems.

(c) As perimeter-radiation-zone valves.

(d) As diverting valves around boilers or cooling towers.

(7) The designer may choose to use either four 2-way valves or one 3-way mixing valve, and one 3-way bypass valve for 2-position flow-control applications as dual-temperature-system changeover valves.

(8) Control valves are classified according to the action of the valve spring in moving the disk or plug relative to the seat when the control signal or the power is removed. A 2-way valve that opens its flow port under this condition is called a normally-open (NO) valve, and one that closes its flow port under this condition is called a normally-closed (NC) valve. A 3-way mixing valve has both NC and NO inlet flow ports connected to a common (C) outlet flow port. A 3-way bypass valve has both NC and NO outlet flow ports connected to a C inlet flow port.

(9) The flow-regulating characteristic of a valve is generally determined by the shape of a disk or plug that passes through the flow port. The flow-regulating characteristics used for control systems covered by this manual are:

(a) Linear flow, in which the percent of valve travel equals the percent of maximum flow rate through the valve.

(b) Equal-percentage flow in which equal increments in the percentage of valve travel produce an equal-percentage change in flow rate from the previous flow rate, when a constant pressure drop is maintained.

(10) The applications of 3-way mixing valves covered by this manual require the use of valves with linear flow characteristics. The applications of 2-way valves covered by this manual require the use of valves with equal-percentage flow characteristics. This requirement results from the application of the valves as modulating fluid-control devices. The equal-percentage-flow characteristic matches the non-linear heat-exchange characteristics of the HVAC equipment coils with a change in fluid flow that tends to linearize the heat-exchange output of the coil with a linear signal to the control valve. The linear-flow characteristic is more suitable for mixing applications and for humidification.

(11) The purchase price of a control valve increases with its size. The installation cost of a control valve also increases with its size, because of:

(a) The change from screwed ends to flanged ends

(b) Because larger valves and their weight require more installation and handling labor.

(12) At a pipe size of 4 inches or larger, a type of rotary control valve (known as a butterfly valve) becomes economically suitable for HVAC control applications because of the combination of the price of the valve and the installation costs. The butterfly valve has a disk that rotates on a shaft and closes against a seat. The seat is concentric with the connected pipe. The butterfly valve can have flow-control characteristics similar to equal percentage when used with an appropriate actuator and positioner. In three-way applications of valves for 4-inch pipe size and larger, the designer will show two valves on a common pipe tee, with separate actuators that will operate the two valves simultaneously. One of the valves will be NC, and one will be NO. The C connection can be either an inlet or an outlet. This allows the combination of 2 valves and a pipe tee to function as a 3-way mixing valve or a 3-way bypass valve. Figure 2-1 shows butterfly valves used in 3-way mixing and 3-way bypass arrangements on a common pipe tee.

b. Control dampers.

(1) Dampers are used to regulate the flow of air in ductwork in both modulating and 2-position control applications.

(2) Examples of the use of modulating dampers are:

(a) Air plenum temperature control by mixing outside air and return air.

(b) Space temperature control by mixing warm air and cool air.

(c) Space temperature control by varying the flow of cool air.

(3) Examples of the use of 2-position dampers are:

(a) Closing outside-air dampers or building-exhaust dampers when fans are stopped.

(b) Isolating sections of ductwork for smoke-control purposes.

(4) Dampers are classified by the action of their blades, which connect to a common shaft that is rotated to open or to close the damper. Opposed blade dampers provide better flow characteristic in throttling applications. A throttling application is one where the damper is installed in series with the path of flow and the damper is used to add pressure drop to reduce air flow. Parallel blade dampers are used to provide better flow characteristics in mixing applications. A mixing application is one where more than one flow path exists in parallel. Usually, two or more dampers are installed in parallel to each other and the dampers divert flow rather than increase total system pressure drop.

(5) The control action of dampers (NC or NO) depends on the direction of their blade rotation caused by the spring-return stroke of an actuator connected to the damper's drive shaft, when the control signal or power is removed.

(6) When a control-system application requires that a damper be open prior to the start of a fan, an adjustable switch is connected to the damper; this device is called an

end switch or limit switch. The end switch operates a set of contacts in the fan-starter control circuit when the damper is fully open, to allow the fan to start; the end switch opens the circuit to prevent the fan from continuing to operate if the damper begins to close.

c. Actuators.

(1) Actuators are used to operate valves and dampers. Pneumatic actuators are powered by air pressure, and are controlled directly by a pneumatic control signal and indirectly by an electric or electronic signal. An electro-pneumatic device converts an electric or electronic signal to a pneumatic signal to stroke the actuator. Electric and electronic actuators are electrically powered and are controlled directly from an electric or electronic signal to stroke the actuator.

(2) Modulating control of actuators requires either the use of a 4 to 20 milliampere control signal directly to an electronic actuator or the conversion of the signal to a pneumatic control signal of 3 to 15 psig. The pneumatic signal can be directly or inversely proportional to the electronic signal. The signal conversion values are shown in figure 2-2.

(3) Two-position control of electric actuators requires the closing and opening of a contact to operate an electric actuator. Two-position control of pneumatic actuators requires an electric/pneumatic device to pass 20 psig main air to the actuator, or to exhaust air from the actuator.

(4) Sequencing occurs when actuators are modulated from a common signal by using a portion of the 4 to 20 milliampere signal or the converted 3 to 15 psig signal. The actuator stroke is adjusted to move its connected valve or damper from fully-closed to fully-open over the assigned portion of the common control signal. Deadbands between the movement of valves and dampers are achieved by assigning a portion of the common control signal as a deadband. Each actuator is adjusted so that its full stroke occurs on either side of the deadband limits outside of the deadband. Examples of the use of sequencing with a deadband are:

(a) Sequencing of heating and cooling with a dead-band between heating and cooling.

(b) Sequencing of heating and outside ventilation air beyond the required minimum quantity with a dead-band between heating and increased ventilation.

(5) Actuators are modulated in parallel by assigning the identical portion of the control signal to each actuator for its full stroke. Modulation in parallel occurs in air-stream mixing applications such as:

(a) Modulation of outside-air, return-air and relief-air dampers for free cooling.

(b) Modulation of multizone hot-deck and cold-deck dampers in parallel.

d. Current-to-pneumatic transducers. The modulating device for converting a current-control signal to a pneumatic-control signal is a current-to-pneumatic transducer (IP). A 20-psig main air supply to the IP is the source that develops a 3- to 15-psig output signal in a scaled relationship to a 4- to 20-milliampere input signal.

e. Solenoid-operated pneumatic valves. The 2-position device for converting an electric-contact closure signal to a pneumatic signal is the solenoid-operated pneumatic valve (EP). The EP is a 3-way valve that connects the normally-closed and common ports when the solenoid coil is energized and connects the normally-open and common ports when the solenoid coil is de-energized. The EP is used to switch 20-psig main air to the actuators and to exhaust air from the actuators.

f. Positive Positioners.

(1) All modulating-control applications of pneumatic actuators require that the actuator be equipped with a positive positioner (PP). A main-air supply is the source of its operating power. The device throttles main air as required to stroke the actuator to the position dictated by the pneumatic control signal. However, the positive positioner can exert pressure higher than that of the pneumatic-control signal and thus can maintain the required position against the opposing force of the HVAC system pressure. Piping-system pressures tend to compress the air in the diaphragm chamber of the valve actuator. The compression causes a shift in the actual operating ranges of the valves. The positive positioner has an adjustable pneumatic-signal start point for the stroke of the actuator and an adjustable pressure span for the full stroke of the actuator. The stroke is proportional to the pneumatic control signal.

(2) Simultaneous heating and cooling can occur when pneumatic actuators are used, even though the spring-operating ranges are selected without an overlap. The results of this phenomenon are shown in figure 2093. Because of this phenomenon, sequencing applications for HVAC systems must have positive positioners on pneumatic valves and damper actuators, to maintain deadbands between actuator operating ranges. A control system with positive positioners is illustrated in figure 2094. When sequencing actuators from a common control signal, the simultaneous use of heating and cooling can accidentally occur if:

(a) Heating and cooling valve operating ranges overlap.

(b) Heating valve and ventilation damper operating ranges overlap.

(c) Heating valve and cooling air damper operating ranges overlap.

g. The choice between pneumatic and electric actuators. All terminal-unit control systems will have electric or electronic actuators. For all other control-system applications, the designer will make an estimate of the total cost of actuators required for all control systems in the project. The designer will take into account the cost of multiple actuators on large dampers and the cost of larger actuators required for higher torques to operate large valves. The total installed-cost estimate of pneumatic actuators will include:

(1) The actuators.

(2) The IPs.

(3) Tubing.

(4) Local indicators.

(5) The cost of the compressed-air system.

The total installed-cost estimate of electric actuators will include consideration of:

(1) The actuators.

(2) Wiring.

(3) Loop-driving circuits as explained in this manual.

h. Existing compressed-air source. If sufficient air is available from an existing temperature-control compressed-air system, it may be used as the air source for additional control systems.

i. Life-cycle cost. After the installed-cost estimates are prepared, a life-cycle cost estimate will determine the choice between pneumatic and electric actuators.

j. Sequencing actuators. The actuators that control valves and dampers are sequenced when HVAC applications require that the process variables be sensed at a common location and controlled from a common modulating signal. The objective of sequencing is to avoid energy waste by preventing the following opposing processes from acting simultaneously:

(1) Heating and cooling.

(2) Humidification and dehumidification.

k. Design requirement in regard to actuator-sequencing ranges. The designer will show the actuator-sequencing ranges in the equipment schedule when standard control signals apply.

(1) Pneumatic actuators are sequenced by connecting the signal input connections of the actuators' positive positioners to the same pneumatic control signal and adjusting the positioners' starting points and spans to achieve the required sequence. For example, two valves can be operated in sequence if their positive-positioner spans are set at 4 psig and their starting points are set at 3 psig and 9 psig respectively. This results in ranges of valve full-stroke operation of 3 to 7 psig and 9 to 13 psig, with a 2-psig deadband between the ranges of operation.

(2) Electric actuators have starting points and span adjustments similar to those of the pneumatic actuator's positive positioner. In this case, the starting points and spans are adjusted in milliampere values. When electric actuators are sequenced, the modulating-control circuit will be designed within a 600-ohm limitation.

l. Multiple actuators connected to the same control damper. When the operating-torque requirement for an HVAC-system damper exceeds the output torque of a single actuator, additional actuators are connected together to operate in parallel to control the damper. The designer is not required to show multiple actuators connected to the same damper on the schematic. The vendor has the information necessary in the contract specification to apply multiple actuators when required.

m. Design of modulating-control circuits within a 600-ohm circuit-impedance limitation.

(1) The output of an HVAC-system controller is connected in series to actuators external to the HVAC control panel, and also to other devices in the HVAC control panel in a direct-current series circuit. The number of devices varies with the complexity of the control sequence, and the impedance of each connected device is additive as a resistance in the circuit. The amount of output-circuit impedance that a controller will tolerate is product-specific. The limitation of 600 ohms in the output-circuit design is needed to permit the controllers of several manufacturers to function in the same circuit. The limitation permits standardization in the design and permits substitution of one manufacturer's controller for that of another during maintenance of the system.

(2) If a modulating-control circuit is designed to use electric or electronic actuators, the impedance can exceed the 600 ohm limitation if:

(a) Multiple actuators are required for the same damper.

(b) More than one damper is modulated from the same control signal, such as in the case of modulating outside-air, return-air, and relief-air dampers.

(c) Multiple control-system devices located within the HVAC control panel are necessary to achieve the sequence of control.

(3) Many of the control-system devices typically add 250 ohms impedance to a series circuit. This 250-ohm impedance value comes from a dropping resistor in the device that is used to convert the 4- to 20-milliampere current signal at 24 volts dc to a 1- to 5-volt signal used by the device's internal circuitry.

(4) Figure 2-5 shows methods for designing circuits within the 600-ohm limitation. The figure shows the following examples:

(a) The limitation exceeded by connecting 750 ohms in series.

(b) Limiting the control-circuit connection to a single actuator.

(c) The circuit designed within the limitation by the use of an additional control-circuit device.

(5) In the first example, shown in the upper part of the figure 2-5, control devices 1, 2, and 3 can be actuators or devices in the HVAC control panel. In the second example, the controller is connected to actuator 1, and actuator 2 is operated by an auxiliary-actuator-driver (AAD) circuit on actuator 1; actuator 2 can operate another actuator by its ADD. In the third example, the same control devices are connected to a loop driver (LD). Control device 1 and the loop driver are connected to the controller. The modulating circuit from the controller is limited to 500 ohms, consisting of 250 ohms for control device 1 and 250 ohms for the input to LD. Control devices 2 and 3 add a total of 500 ohms to the output circuit of LD. The output signal of LD varies in a 1-to-I ratio with its input signal.

(6) Some of the control devices necessary to implement the control sequence have an input impedance of 250 ohms, and their output circuits can accept from 800 ohms to 1000 ohms of impedance. The amount of allow-

able impedance in their output circuits is product-specific. Any control device whose modulating-control output circuit has greater impedance-loading capability than the impedance of its modulating-control input circuit can function as a loop driver. In addition to performing a specific control-sequence function. This output-driving capability is found in most modulating-control devices.

(7) When the control system requires more than one damper with electric or electronic actuators to be modulated by a control circuit, the designer will show the signal (on the schematic) connected to one of the damper actuators. The AAD circuit of that actuator will be shown as connected to drive a separate actuator on another damper, which, in turn, can drive another actuator on still another damper.

(8) When a modulating-control circuit must drive multiple panel-mounted control devices, the designer will show on the schematic:

(a) Not more than 2 devices (such as IPs) connected to that circuit, unless one of the devices is a control device that accepts a modulating input signal and produces a modulating output signal.

(b) Not more than 2 panel-mounted control devices connected to the modulating output of a panel-mounted control device.

(9) The schematic is not intended to show the physical connections to the devices, but rather to show the relationship of the necessary control devices in the control loop.

n. Transmitters. Variables such as temperature, pressure, and relative humidity are sensed by means of elements that are connected to the control loops via transmitters. The output signal of the transmitter is the standard 4 to 20 milliampere dc signal, which is factory-calibrated for zero point and span relative to the input resistance value of the sensing element. The transmitters are 2-wire, loop-powered (i.e., powered by the control-panel power supply) devices that connect in a series circuit with the controller input. The impedance limitation of the circuit in which the transmitter can function is product-specific. A typical value is 700 ohms at 24 volts dc.

o. Single-loop digital controller.

(1) A standard controller, known as a single-loop controller, will be used for all control-system applications except those required for the control of HVAC-system terminal units. In all applications where it is used, the controller will be mounted in a HVAC control panel. The controller-mounting dimensions will conform to a standard panel cutout requirement. The controller will be used for the following applications:

(a) As a controller for maintaining temperature, relative-humidity, static-pressure, and/or airflow setpoints.

(b) As an economizer-mode switchover controller that determines whether outside air is available for cooling.

(c) As an outside-air-temperature controller for scheduling hydronic-heating supply temperature and for starting and stopping pumps.

(2) The controller will be a microprocessor-based device with manually-configurable control features resident in solid-state electronic-memory components. Manual access to the features of the controller will be through a keypad and an alphanumeric indicator on the face of the controller. The controller will have standard features that will allow it to serve all functions prescribed in its application. Standard features will allow the replacement of any controller by a spare not necessarily of the same manufacturer.

(3) The single-loop digital controller will have the following inputs and outputs:

(a) A process-variable analog input (PV).

(b) A remote-setpoint analog input for control-point adjustment (CPA).

(c) An analog output (OUT).

(d) A process-variable-actuated contact closure output (PV contact).

(e) A contact output that responds to the difference between PV and CPA analog inputs (DEV contact).

(4) Some controller features are configurable by manual input, and others are selectable by setting switches or jumper wires. The configuration methods vary with the specific manufacturer and model. The controller will have the selection of features for its inputs and outputs as shown in table 2-1.

Table 2-1
Controller Features

Input	Output	Feature
PV		Scalable to the range of the input transmitter.
CPA		Adjustable bias of setpoint relative to the PV input range.
CPA		Adjustable ratio of setpoint relative to the PV input range.
CPA		High and low limits of setpoint adjustment.
	OUT	Selection of direct-acting or reverse-acting.
	OUT	Adjustable high and low limits.
	OUT	Adjustable bias (Manual reset).
	OUT	Selection of manual or automatic control.
	OUT	Selection of proportional (P), proportional plus integral (PI), and proportional plus integral plus derivative control modes (PID).
	OUT	Anti-reset windup.
	OUT	Selection of manual tuning or self tuning.
	OUT	Manual reset feature for use when the controller is configured from proportional mode only and is manually tuned.
	PV Contact	Adjustable contact setpoint.
	PV Contact	Adjustable hysteresis or deadband.
	DEV Contact	Adjustable contact setpoint.
	DEV Contact	Adjustable hysteresis or deadband.

(5) The output control modes that can be used in combination are:

(a) Proportional mode, which varies the output proportionally to the error between the PV input value and the controller setpoint.

(b) Integral mode, which modifies the output signal as a time-related function of the error between the PV input value and the controller setpoint.

(c) Derivative mode, which modifies the output signal as function of the rate of change of the error between the PV input value and the controller setpoint.

(6) Each of these control modes has an assignable constant, which is adjusted in the process of tuning the controller.

(7) The integral mode will have a feature that automatically stops the integration of the error signal when the controller output signal reaches its minimum or maximum value. This feature is known as anti-reset windup. If the process being controlled does not respond to the controller output for a period of time (such as an overnight shutdown), continuing the integration of the error signal during that period would result in the controller failing to respond to control the process immediately on startup. Without the anti-reset windup feature, the controller would reverse the integration process on startup, but the reversal would require a time period equal to the time period of the integration during shutdown.

(8) The controller will have the following selectable modulating-control functions:

(a) Self tuning of control-mode constants.

(b) Manual tuning of control-mode constants.

(9) Self tuning allows the controller to select the optimal combination of proportional, integral, and derivative control-mode constants. The controller continues to use these selected constants until the self-tuning control function is again selected. Applications where self tuning is used are as follows:

(a) Mixed-air temperature control.

(b) Heating-coil temperature control for multizone hot-deck and dual-duct hot-duct applications.

(c) Modulating-value and modulating-damper preheat-coil control.

(d) Cooling-coil discharge-temperature control for multizone cold-deck, dual-duct cold-duct, and VAV-system discharge temperature.

(e) Relative-humidity control.

(f) Supply-duct static-pressure control.

(g) Return-fan volume control.

(h) Hydronic heating supply-temperature control.

(i) Hydronic-heating space-temperature control.

(10) If integral and derivative control modes are not appropriate for an application, the controller is manually tuned for proportional-mode control, and the integral and derivative mode constants are set to zero. Applications where manual tuning is used are as follows:

(a) Relative-humidity high-limit control.

(b) Outdoor-air scheduling of hydronic-heating supply temperatures.

(c) Space-temperature control of single-zone HVAC units and heating and ventilating units.

(11) When the single-loop digital controller is manually tuned to operate as a temperature controller in the proportional mode, the proportional-band setting is determined by the use of function 2-1.

PB $(T \times 100) / T_s$ (eq. 2-1)

Where:

P_B = Proportional-band constant (percent)
T = Portion of the transmitter span required for full-scale controller output change (degrees F)
T_s = Temperature span of transmitter (degrees F).

(12) If the controller output is to change from 4- to -20 milliamperes over the range of 0 to 60 degrees F and the transmitter range is in the range of minus 30 to plus 130 degrees F (a span of 160 degrees F), the use of equation 2-1 would result in:

PB = 37.5 percent.

(13) When a single-loop controller is configured for proportional control mode only, the output of the controller must be set to match the value required for the application when the PV input value coincides with the controller setpoint. This is accomplished by configuring the manual reset setting as a percent of controller output. The effect of this configuration parameter is to shift the controller output throttling range with respect to the controller setpoint. As an example, refer to figure 2-6 which shows the setpoint relative to the controller throttling range for manual reset settings of 25, 50 and 75 percent for a direct-acting controller.

(14) When used as an economizer-switching controller, the functions of the controller inputs and outputs will be as follows:

(a) A return-air temperature transmitter connected to the PV input.

(b) An outside-air temperature transmitter connected to the CPA input.

(c) An output contact configured as PV and acting as a switch.

(d) An output contact configured as DEV and acting as a switch.

(15) As the outside-air temperature changes, the potential for using outside air for cooling is affected. When the outside-air temperature is lower by a specified number of degrees than the return-air temperature, outside air can be used for cooling; when the reverse is true, outside air must be at minimum quantity. The PV contact setpoint prevents the use of outside air beyond minimum quantity until the return-air temperature rises to its set-point. When the return-air temperature rises to the PV contact setpoint, the HVAC system is no longer experiencing a heating load. The DEV contact setpoint allows the use of outside air until the outside-air temperature approaches the return-air temperature. The DEV contact then puts the dampers under control of a minimum-position switch. A minimum-position switch is a device whose control output is manually modulated to an actuator that then remains fixed until reset manually. Both conditions must be met PV and

DEV contacts closed) for the use of outside air beyond minimum quantity.

p. Function Modules.

(1) There are control functions required with less frequency in HVAC control applications than those included in the prescribed version of the single-loop controller. Control devices to perform specific control functions not available in the single-loop digital controller are called function modules in this manual. Function modules will be located in the HVAC control panel, except as noted. Function modules accept contact, analog, or gradual manual adjustment input signals to provide:

(a) Signal selection,

(b) Signal inverting,

(c) Contact transfer output from analog signal input,

(d) Contact transfer output from the comparison of two analog input signals, and

(e) Generation of constant analog-output signals.

(2) A minimum-position switch is a manually-adjustable modulating output device used to hold an outside-air damper open to admit minimum ventilation air. This same device is used as a temperature-setpoint selector. In this application, it allows manual adjustment of the CPA input signal to a single-loop digital controller. In the case of certain single-zone HVAC systems, this device is wall-mounted and accessible to the occupant for adjustment of the space-temperature controller setpoint.

(3) A signal invertor is a modulating-input and modulating-output device used to reverse the direction of its input. The signal reversal is required when spring-return position of a valve or damper actuator has been chosen to operate with a direct-acting signal, and when an actuator with which it must be sequenced has been chosen to operate with a reverse-acting signal. For example, a chilled-water coil is used for dehumidification and cooling. The chilled-water-coil valve is chosen as NC because it is to be sequenced with a NO heating-coil valve, and is to close when the control signal is removed during fan shutdown and during the unoccupied mode. The direct-acting temperature-control signal is correct for this combination. The humidifier control valve is also chosen as NC because it also is to close when the control signal is removed on fan shutdown and during the unoccupied mode. The humidity-control signal must be reverse-acting when it operates the humidifier valve, and must be direct-acting when it operates the cooling-coil valve. A signal invertor is used to reverse the signal direction of the humidity controller signal to the cooling-coil valve.

(4) A signal selector is a device with multiple modulating inputs and a modulating output. This device is used to select the highest or the lowest of its input signals as its output.

(5) A sequencer is a modulating-input and time-delayed contact-output device which operates its outputs in sequence and is used in applications that require the operation of stages in refrigeration control from a modulating-control signal. The time delay prevents all stages of refrigeration from starting simultaneously when the HVAC system starts. The contact opens instantaneously on power failure (or on signal failure) to the sequencer. This instantaneous action prevents all stages of refrigeration from starting simultaneously if electric power to the control system is interrupted momentarily.

(6) A loop-driver module is used where required to allow modulating-control circuits to be designed within a 600-ohm impedance limitation. A loop-driver module can be any modulating-input or modulating-output device used for this purpose alone or while performing an additional control loop function. Signal selectors and signal invertors strategically located in control loops often perform such dual service.

q. Relays, including time-delay relays. All relays, including time-delay relays, will be 2-pole, double-throw devices; they are used for control-system interlocking functions and will be located in the system's HVAC control panel.

r. Time clocks.

(1) A time clock will be used to control the timing of the modes of operation of an HVAC control system when the control system is not interfaced with EMCS. When a time clock is used, it will be located in a HVAC control panel. The modes of operation are occupied, unoccupied and ventilation delay.

(2) The time clock will be a device that accepts a time schedule by manual input through a keypad and an alphanumeric display. The time clock features will be:

(a) Two independently-time-controlled contacts.

(b) A program of 4 "on" events and 4 *"off'* events for each contact.

(c) 365-day schedule.

(d) Twelve selectable holidays.

(e) Standard-time and daylight-time adjustments.

(f) Timed override of programs.

(g) Battery backup of memory.

(3) When used to time the modes of operation of air handling systems, one contact of the clock will be used for occupied and unoccupied timing; the second contact will be used for ventilation-delay mode timing. For other applications, the contacts may be used as convention to the design.

2-4. Control loop

A control loop performs three distinct functions: sensing of a variable as the input to a controller; decision-making or control based on the valve of the input; and output or actuation as a result of control. Figure 2-7 illustrates a simple control loop. The input signal is a continuous analog of the process, and the controller either continuously sees the input or continually scans it The controller changes its output as required by changes in its input.

2-5. Open control loop

When a control loop senses a variable, makes a control decision, and sends an output signal to a control device without receiving input information related to the results of its control action, the control loop is said to be an open loop. There are some open-loop control applications used in HVAC control, such as:

(1) Operation of pumps above or below a certain outside temperature.

(2) Automatic stopping of HVAC systems based on outside-air temperature.

(3) Scheduling of hydronic-heating supply temperatures based on outside temperatures.

(4) Timing and time-delay operations. Figure 2-8 illustrates an open control loop.

2-6. Closed control loop

When the controller changes its output decision based on updated input information, the control loop is said to be a closed loop. Most of the control loops used in HVAC control are closed loops. Control of coil-air discharge temperatures is an example. The transmitter, connected to a temperature-sensing element in the air stream passing through the coil, signals the temperature controller; the controller makes a decision as to whether to open or close the valve that allows water to flow through the coil; and an actuator operates the valve. The feedback in this example is the continuous input to the controller of a changing temperature signal from the coil-air discharge temperature sensor and transmitter. The transmitter continuously updates the controller on temperature information from the sensor, and the controller modifies its output to control the valve. See figure 2-9 for an example of a closed control loop.

2-7. Application of open-loop control and closed-loop control to HVAC Systems

a. Open loops and closed loops in combination. Open loops and closed loops are used in combinations in some HVAC control-system applications. A perimeter hydronic-heating system may have open-loop components to start and stop the pump and to schedule the supply-water temperature based on outside-air temperature. At the same time, it may have a closed loop for the control of the supply-water temperature.

b. Closed loops in combination. There are some HVAC control applications that use two simultaneously-acting, closed control loops to actuate the same device. An example, as shown in figure 2-10, is control of a duct humidifier. A space relative-humidity transmitter is the primary input to a relative-humidity controller for the humidifier valve in a closed loop (1oop 1). A duct-mounted humidity transmitter downstream of the humidifier signals a high-limit relative-humidity controller, which provides a high-limit closed-loop control function (loop 2), by overriding the primary controller loop 1 to shut off humidification by closing the valve if the relative humidity of the air stream rises to the loop-2 relative-humidity high-limit setting.

2-8. Typical control modes

a. Two-position control.

(1) Some HVAC equipment can be turned on and off as a method of temperature control. This type of HVAC equipment is not applied where temperature control between close limits is required.

(2) When a thermostat or other control device cycles equipment to maintain its setpoint the control mode is called two-position control. A thermostat used for two-position control opens and closes contacts for control rather than providing a modulating output signal. The contacts either open or close when the temperature is at the thermostat setpoint. The state of the contact reverses when the temperature changes in the proper direction. Such thermostat contacts usually either open on a temperature rise (in a heating application), or close on a temperature rise (in a cooling application). The temperature at which this happens depends on the switch temperature differential (hysteresis).

(3) An example of two-position control is unit-heater control, in which a $_{space}$ thermostat turns on a unit heater when the space temperature drops to 65 degrees F and turns it off when the space temperature rises to 67 degrees F. The thermostat is said to have a differential of 2 degrees F and a setpoint of 65 degrees F. This type of control can result in a slight undershoot below the lower end of the differential, and a slight overshoot above the higher end of the differential.

(4) The thermostat may turn off the unit at 67 degrees F, but the heating load may decrease due to increasing outside-air temperatures. In this event, water circulating through the unit coil, which will then be acting as a radiator with the fan off, may raise the temperature in the space slightly above 67 degrees F as an apparent overshoot. Even though the unit turns on when the space temperature drops to 65 degrees F, the space temperature may fall slightly below 65 degrees F after the unit starts. This depends on the heating capacity of the unit. If the heating load decreases, the temperature may subsequently rise to 66 degrees F, and stay at that temperature for a considerable time, while the fan continues to run. A similar situation can happen after the unit heater shuts off at 67 degrees F and the space temperature drops to 66 degrees F. Consequently, the space temperature may be 66 degrees F with the unit heater fan either running or stopped. A graphic representation of two-position control is shown in figure 2-11.

b. Modulating Control. A simple control loop is shown in figure 2-12 as it would be applied to heat outside-air for ventilation using a pneumatic-valve actuator rather than an electric- or electronic-valve actuator. The controller operates an IF in response to the signal of the temperature-sensing element in the air duct, downstream of the coil, via a transmitter. The IP pneumatic output signal modulates the positioner on the pneumatic valve actuator. The positive-positioner output throttles main air to the actuator, which moves the valve stem. This example is used to explain two modes of modulating control that are applicable to the control of valves, dampers, inlet vanes, and other devices. The modes applicable to most HVAC-control applications are:

(1) Proportional mode (P).

(2) Proportional plus integral mode (PI).

c. Proportional mode (P). The most common control mode in HVAC control is proportional mode.

(1) Proportional mode is used for the following applications:

(a) As a method of scaling an outside-air temperature signal to schedule water temperatures for heating.

(b) As a method of space-temperature control for single-zone air-handling units.

(c) As a method of controlling terminal units that can be modulated.

(2) Figure 2-13 shows the kind of control that would be expected if the controller in figure 2-12 were configured for the proportional-control mode. The controller modulates its output signal in proportion to variations of the input signal. For example, the controller, operating through the IP, sends a 3-psig air signal to the normally-open preheat-coil valve when the discharge temperature is 43 degrees F, and it sends a 15-psig signal when the discharge temperature is 47 degrees F. The 3-psig signal completely opens the valve to heating, and the 15-psig signal completely closes the valve to heating. The controller/IP combination has a proportional sensitivity of 3 psi per degree F, and throttles the valve over a range of 4 degrees F. The setpoint of the controller is 45 degrees F, but the temperature at which it is controlling is somewhere between 43 degrees F and 47 degrees F. It takes a 4-degree F change in controller-output signal for the valve to go from full heating to no heating.

(3) On a fall in outside-air temperature, the heating load increases. The resultant drop in temperature at the sensor causes the transmitter to signal the controller to lower its output to the IP transducer. The IP transducer lowers the control air pressure to the normally open valve, allowing more heating medium to pass through the coil, thereby returning the discharge temperature almost to the temperature maintained before the fall in outside temperature. Increasing the controller's sensitivity would shrink the full heating/no heating temperature range; decreasing its sensitivity would increase that range.

(4) The phenomenon that prevents the proportional-control mode from achieving its control point exactly at setpoint is called "offset due to loan". This occurs when an equilibrium is reached between HVAC-system output and the load imposed on the HVAC system. When this equilibrium occurs, the discharge temperature does not change. This means that the proportional-mode controller does not change its output and cause a change in the HVAC-system output. Conversely, without the change in HVAC-system output to drive the temperatures toward the controller setpoint, the controller output does not change. Therefore, at these equilibrium points, the controls do not bring the system back to setpoint.

(5) Figure 2-13 shows the relationship between coil discharge temperature and controller/IP output in response to changes in outside-air temperature. Except between point A and B the controller/IP output is proportional to the discharge air temperature. Between points A and B, the coil discharge air temperature is the same as the outside-air temperature. At points A and B, with a coil discharge temperature of 47 degrees F, the output is 15 pounds and the valve is closed. At the fifth hour when the discharge temperature is at the 45 degrees F setpoint the controller/IP output is 9 pounds and the valve is half open. If the valve is closed at 47 degrees F outside-air temperature and is half open with a twenty-degree pickup at 25 degrees F outside-air temperature, then by extrapolation the valve is fully open at outside-air temperatures of 3 degrees F and below. As soon as the air temperature leaving the coil, the incoming outside-air temperature, and the heating action of the coil reach an equilibrium, the valve remains at a given position until the equilibrium is disturbed by changes in the outside-air temperature.

(6) For a given load on the system, there is some optimal proportional-sensitivity adjustment at which the controller will be in stable control and close to the setpoint. In some HVAC control applications, proportional control may function quite well with a high-sensitivity adjustment. A high-sensitivity adjustment results in a narrow range of temperature drift from setpoint while the controller changes its output from full output to minimum output. Too high a sensitivity adjustment causes the control point to continuously overshoot and undershoot the setpoint. In other control applications, stable control may not be achievable with a high-sensitivity adjustment. A low-sensitivity adjustment results in a wide range of temperature drift from setpoint while the controller changes its output from full output to minimum output, but control is usually more stable. After the controller sensitivity has been adjusted, changing conditions of load due to seasonal and other factors tend to make the adjustment less than optimal. This phenomenon is a function of HVAC-system capacity and HVAC-system response to load changes. The sensitivity of a proportional controller to process variable changes is called proportional gain.

(7) As long as the proportional controller is controlling in a stable fashion, at varying load conditions, at some control point near the setpoint and the proportional-gain setting is optimum, the controller has achieved the most precise control of which it is capable.

d. Proportional plus integral mode (PI).

(1) Many control applications require a controller that can eliminate offset due to load and can control very close to setpoint. To achieve closer control than is available from the proportional-control mode, some automatic adjustment has to be made to the controller output to change the actuator position without changing the setpoint or the proportional-gain setting. The method used to adjust the controller output for changes in load is called "integral mode". Integral mode adds a gain component algebraically to the controller output. This component is time-proportional to the difference between the setpoint and the stable control point produced by the controller's proportional gain. This difference is caused by the offset due to load and is called steady-state error, which is the error between control point and setpoint when a balance between the load on the system versus the system capacity output and controller output is established. Steady-state error differs from the transitory error between set-point and control point due to an upset in the process, such as a changing load or a step change in setpoint.

(2) The integral mode adds a component of output to the output of the controller that is produced by the controller's proportional gain. The size of the component is determined by the integral gain multiplied by the error. As the error decreases, the size of the component integrated to the output signal also decreases, and becomes zero when the controller is controlling at the set-point. This added component of output causes the valve-actuator-stroke position to change. This change in valve capacity resulting from the change in the actuator-stroke position upsets equilibrium or prevents equilibrium from occurring until the control point reaches setpoint. This action causes two things to happen. The resulting temperature change due to the change in valve-actuator-stroke position causes the proportional gain to change its component of output signal at a magnitude proportional to the change in the temperature being sensed. While this is happening, the error changes because the control point is closer to setpoint, and this in turn causes the integral gain to make a change in the magnitude of its component of the output signal due to the changing value of the magnitude of the error between control point and setpoint. There will be no further change in the proportional-mode or integral-mode output components when the steady-state error is zero. Because of the combined action of both of these control modes, the controller can reduce the offset to zero, or nearly zero, and can establish a steady-state equilibrium of HVAC system control at a value very near setpoint. This type of control is called Proportional-Integral (PI) control.

(3) See figure 2-14 for an illustration of proportional plus integral control modes. The upper graph of the figure is identical to the upper graph of figure 2-13. The lower and middle graphs provide a comparison of the proportional-mode controller output and resulting coil-air discharge temperatures (light lines) versus those of a PI-mode controller (dark lines). In the beginning of the middle graph the integral-mode component is positive and is added to the proportional mode component. This additional pressure closes the normally-open valve where the outside-air temperature reaches setpoint 45 degrees F, instead of at point A, at which the coil discharge-air temperature is 45 degrees F plus half the controller throttling range or 47 degrees F (45 $ 4/2). The 47 degrees F temperature would have occurred with proportional control, at point A. Similarly, when the outside-air temperature falls to 47 degrees F. the integral-mode component prevents the valve from opening until the outside-air temperature falls below 45 degrees F. The proportional only mode controller would have begun to open the valve at 47 degrees F outside-air temperature at point B. From the time the valve begins to open at the outside-air temperature of 45 degrees F until the fifth hour when the outside-air temperature is 25 degrees F the integral-mode component of the controller output signal becomes smaller until at the fifth hour the integral-mode component becomes zero. This additional pressure on the normally-open valve allows the coil air discharge temperature to remain at the 45 degrees F setpoint instead of controlling between 47 and 45 degrees F when the outside-air temperature is between 45 and 25 degrees F as would have occurred with proportional only control. Similarly, when the outside-air temperature falls below 25 degrees F, the integral-mode component is subtracted from the proportional-mode component of the controller output signal to open the valve enough to keep the discharge-air temperature at 45 degrees F rather than at 43 degrees F as would have occurred with proportional-mode control.

e. Effects of rapid load changes. The rate of change of load imposed on the HVAC system by the process affects how well the controller will perform its task of controlling at setpoint. The temperature of the outside air changes relatively slowly, and the temperature conditions inside also require some time to change. Inside conditions change as a function of air-temperature changes made by the HVAC system, which warm up or cool down masses of material within the building. Inside conditions also change as functions of lighting load and occupancy. Because of these relatively slow rates of change, most of the HVAC processes that require gradual controller output changes can be controlled quite well with proportional plus integral (Pi) control modes. Except for lighting loads on the HVAC system, these variable changes are relatively slow compared to the rates of change of variables that affect some non-HVAC processes. Lighting loads are sometimes imposed on the HVAC system quickly. This is an example of a step change in the process variable. The combined actions of proportional and integral modes are not always adequate to control rapidly-changing variables.

f. Proportional-integral-derivative (PID) mode.

(1) Some processes require a controller that can respond to rapidly-changing process variables. One answer to control of rapidly-changing processes has been the addition of another control mode called derivative mode. When this control mode is added to proportional-integral control, the combination is known as proportional-integral-derivative (PID) control mode. The PID control mode adds a component algebraically to the output signal; this component is proportional to the rate of change of the error between the control point and setpoint. This automatic adjustment also affects the proportional and integral output components in a manner analogous to the way in which the integral component affects the proportional component. As the valve-actuator-stroke position changes, the temperature changes as a result of changing flow through the valve, and the rate of error-signal change between the control point and setpoint varies as the control point comes closer to setpoint.

(2) There are a few HVAC-control applications that are difficult for either P-mode or PI-mode control because of the fast rates of change of the process variables. One such application is the control of tankless hearing converters, such as might be found in some domestic hot-water-heating applications. The I-mode component of Pi-mode takes care of the varying range of offsets due to loads that occur in domestic hot-water heating applications. For example, high-rise residential buildings have morning and evening peak periods of demand for hot-water use. These peak-demand periods drive the domestic hot-water temperature to the low end of the offset range. Periods of relative nonoccupancy, such as late morning and early afternoon, drive the temperature to the high end of the offset range due to the minimal demand for domestic hot-water use. Periods of relative inactivity during occupancy, such as late evening and very early morning, require practically no domestic hot water. It is this period that defines the top end of the offset temperature range. The P-mode control alone does not control the water temperature very close to the controller setpoint in this kind of application. The addition of the I mode to the P mode makes the offset range much narrower than would occur with P mode alone. However, Pi modes alone cannot handle the unpredictable diversity of demand as the peak periods start and end. What happens during the period of light demand for hot-water use is that the turn-on of a shower or the startup of a dishwasher produces an upset in equilibrium that has a greater effect than the same event would have during a period of heavy demand. Periods of heavy demand tend to filter out some of the effect of a single turn-on.

(3) In the control-loop applications where manual tuning is prescribed, the proportional-mode constant is set as the result of a calculation. These applications cannot use the integral or derivative control modes. Self tuning is prescribed for these applications because finding the optimum settings manually is difficult and time-consuming. When controllers self-tune, these settings are automatically optimized, and an optimized derivative-mode setting is selected due to the controller's self-tune feature. The effects of adding the I and D modes to the P mode is illustrated in figure 2-15, which shows the results that would be expected with a step change in set-point. Step changes in setpoints rarely occur in HVAC control system applications. The illustration of the step change in setpoint is used to graphically explain the actions of P, I, and D modes.

2-9. Standard HVAC-system control loops

a. Standard HVAC-system control loops will be used for the design of HVAC-control systems. The standard HVAC-system control loops use the single-loop digital controller and additional components. These components are collectively called the control-loop logic. The logic varies with the loop requirements, and its purposes are to interface the loops with the operational-mode signals, to modify signals, and to interface with EMCS. The control-loop logic is implemented by the use of combinations of relays and function modules.

b. Relay coils are activated by occupied-unoccupied, ventilation-delay, safety-shutdown, EMCS-override, and other signals external to the HVAC-control system. The contacts of the relays interrupt analog signals of controllers and function modules, provide inputs to on/off control loops (such as starter circuits), and operate HVAC control-panel pilot lights.

c. All the relays, contacts, and function modules are defined for each loop. The relative physical locations of the relays and function modules will be assigned in the HVAC control panels.

d. Each of the standard HVAC control loops is described in this technical manual as the designer will show them on the contract drawings. Each control loop on the drawings will show the elements of the control loop, such as sensor-transmitters, controller, function modules, relay contacts, current-output to pneumatic signal converter (IP), and final actuator. The control loops will show all the field-mounted and panel-mounted devices for a standard loop. Each control loop will show function modules and relay contacts that are defined for the interfacing of analog control loops with on/off control loops.

2-10. Sizing and selection of control-system devices.

a. The designer will estimate the required motor horse-power of the HVAC control-system air compressor, to provide the proper requirements for incorporation into the electrical power design.

b. Control-system devices such as dampers and valves are sized on the basis of capacity requirements and allowable pressure drops and velocity ranges. The selection of the type of valve or damper is based on factors such as allowable leakage rates and available or practical size ranges.

c. The designer will analyze piping-circuit pressure drops and their effects on the system's pressure drop due to the control valves, based on criteria presented in this manual. The designer will calculate the required liquid-flow coefficients © for each valve and will show the C_v for each valve. The designer will make sure that pumps are selected to include the pressure drop through the total circuit, and will then show the pressure against which the valve must close.

d. The designer will size control dampers based on the criteria presented in this manual and will show the size of each damper.

e. Oversizing of control devices would result in systems in which it would be difficult or impossible to obtain satisfactory control-loop operation, regardless of the quality of the controller and components used. The designer must not assume that self-tuning controllers will compensate for oversizing of control devices. The I and D modes compensate for HVAC-system load variations and HVAC-system equilibrium upsets, but do not compensate for incorrect valve and damper sizing.

2-11. Sizing of the air-compressor motor

a. Calculation. The designer will estimate the required air-compressor-motor size, using equation 2-2, in order to coordinate with the power circuit serving the air compressor.

$$Tr = (Qc \times 100) / Qd \qquad (eq.2-2)$$

Where:
Tr = Air-compressor running time (percent).
Qc = Control-system air consumption in standard cubic inches per minute (scim).
Qd = Air compressor delivery in standard cubic inches per minute (scim).

b. Running-time criteria. The designer will use the following running-time criteria for sizing air-compressor motors:

(1) *New air compressors.* The running-time design criterion for new air compressors is that the running time will not exceed 33½ percent. This requires that, initially, the delivery capacity of the air compressor must be at least three times the estimated air consumption for the whole control system that it serves. The designer will count the number of each type of air-consuming device shown on the schematics and apply their characteristic consumption values to arrive at the consumption total. The designer will base the calculation on the air consumption for each device as shown in table 2-2.

(2) *Existing air compressors.* Control systems can be added to an existing air compressor until the running time reaches 50 percent.

Table 2–2
Air Consumption of Control Devices

Device type	Air consumption (scim)
IP	435
Butterfly-valve actuator positive positioner or inlet-vane actuator positive positioner	865
Damper or valve positive positioner	10
EP	0

c. Application of air-consumption values to sizing of air-compressor motors. An example of the application of the values in table 2-2 is as follows:

5 IPs @ 435 scim each	= 2,175 scim
15 valve and damper positioners @ 10 scim each	150 scim
	2,325 scim

Therefore,

2,325 x 3 (for **a** running time) = 6,975 scim

A check of manufacturers' catalog shows that a 1-hp compressor delivers about 6,560 scim and that a 1½-hp compressor delivers about 8,900 scim. Thus, a 1-hp compressor for the indicated control system would run 35 percent of the time, while a 1½-hp compressor would run 26 percent of the time. On the basis of this estimate, a power circuit capable of serving a 1½ hp motor would provide adequate margin to compensate for variances in the air consumption of control devices selected by the control-system vendor.

2-12. Determination of control-valve flow coefficient (C_v).

a. The control valve flow coefficient (C_v) is a number representing the number of gallons of 60-degree F water that will flow through a valve with a 1 psi pressure drop. The designer will calculate and select the liquid-flow coefficient (C_v) of all modulating-control valves required for the design, and show the pressure against which the valve must close. The selection of the valve's C_v provides the guidelines for the vendor's selection of the valve's port size. The close-off pressure information gives the required criterion for sizing the actuators for the valve selected by the vendor.

b. A valve that is operated in a 2-position open and closed manner will be a line-size valve with the largest available C_v, in order to reduce the pressure drop across the valve and the pumping horsepower required. This applies to dual-temperature-system changeover valves and dual-temperature fan-coil-unit 3-way valves.

c. The designer will select the C_v of the valve based on the maximum flow and the pressure drop for the valve. The selected control valve will be checked against manufacturer's catalogs to ensure that such a valve C_v is available in a product of the control-valve manufacturers.

d. To ensure good control characteristics, the pressure drop through the control valve at full flow must be greater than the pressure drop through the coil and piping circuit (without the control valve) between the point where the piping circuit connects to the supply and return mains. The pressure drop through the control valve will then be at least 50 percent of the total pressure drop through the circuit. The valve C_v must be specified accordingly. For liquid service, the vendor may supply a control valve with a plus 25 percent deviation from that specified. The designer must check to ensure that the pressure drop through the valve will be acceptable at both the specified C_v and at a C_v 25 percent greater than specified. As the C_v increases the pressure drop through the valve decreases. For steam service, the vendor must supply a control valve with a C_v not less than the value specified and not larger than the manufacturers next larger value.

2-13. Calculation of liquid-control-valve flow coefficient (C_v)

a. A physical phenomenon known as cavitation is a cause of valve failure. Cavitation is caused when the velocity through the valve creates an absolute pressure lower than the vaporization pressure of the liquid. To avoid cavitation, use equation 2-3 to determine the maximum allowable pressure drop through open valves.

$$Pm = Km (Pe - Pv) \qquad (eq. 2-3)$$

Where:
Pm = Maximum allowable pressure drop through the valve (psid).
Km = Valve pressure-recovery coefficient (use 0.45).
Pe = Absolute pressure entering the valve (psia) (by design calculations).
Pv = Absolute vapor pressure of the liquid (psia) (from steam tables).

For examples: If 200-degree-F water with an equivalent vapor pressure of 11.53 psia is to flow through a valve with an inlet pressure of 20 psig (34.7 psia), the maximum allowable pressure drop through the valve, as a result of using equation 2-3, will be 10.43 psid.

b. The pressure drop through the control valve must be at least *50* percent of the total pressure drop, as full flow, through the circuit. Continuing with the above example, assume a coil with a flow of 50 gpm and a 4 psig drop through the coil, piping and fittings between the mains. The maximum allowable pressure drop to avoid cavitation was calculated to be 10.43 psid. An initial valve pressure drop selection of 6 psid will prevent cavitation and provide sufficient valve pressure for good control. The C_v can be calculated from:

$$C_v = \frac{Q}{[P/G]} \, {}^{1\!/_2} \qquad \text{(eq. 2-4)}$$

Where:
C_v = Liquid-valve flow coefficient (gpm per psid)
Q = Gallons per minute (gpm).
P = Pressure drop through valve (psid).
G = Specific gravity of the liquid.

The specific gravity of water is 1. As the result of using equation 24, C_v = 20.41. If the designer specifies a of 20, the vendor can provide a valve with a C_v in the range of 20 to 25. If the vendor provides a valve with a C of 25, the pressure drop would be 4 psid as calculated by using equation 24. This valve selection would make the pressure drop through the valve exactly 50 percent of the drop through the total piping circuit. Knowing that a plus 25 percent deviation in valve C_v selection is allow-able, the designer could select a C_v which would result in a larger pressure drop through the control valve. With a C_v of 16, the valve pressure drop will be 9.76 psid as calculated by the use of equation 24. This is less than the maximum allowable valve pressure drop of 10.43 psid, and the circuit pressure drop would be 4 psid (coil and piping) plus 9.76 psid (valve) = 13.76 psid. A C_v of 16 would result in the pressure through the valve being 71 percent of the drop through the piping circuit. If the designer specifies a C_v of 16, the vendor can provide a valve with a C_v in the range of 16 to 20. Cavitation will not be a problem in this example if the water temperature used in 200 degrees F, or less. In most HVAC control-valve applications, water temperatures are less than 200 degrees F. When the hot-water temperature is scheduled from outside air, 200-degree-F water will occur only on design heating days. Also, cavitation should not be a problem in hydronic systems when expansion tank pressures are adjusted so that cavitation is not present at the pump inlet. Selection by the designer of a control valve C_v of 20 rather than a C_v of 16 would be technically acceptable according to the criteria of this TM, but would not be a good engineering practice. If the larger C_v is used as the basis of design, the contractor may provide a valve with a C_v of 25. A C_v of 25 may result in the valve pressure drop being less than 50 percent of the drop through the circuit. The basis of design

valve C_v is selected because of the drop expected through the basis of design coil and the piping system configuration shown by the designer. The coil provided by the contractor, field piping conditions and aging of the piping system are factors which could result in the installed valve being less than 50 percent of the drop through the coil when the contractor provides a valve with the largest C_v permissible.

c. Review of manufacturers' catalogs shows the ready availability of valves with C_v in the range of 16 to 25, as listed in table 2-3.

Table 2-3
Available Control Valves with C_v in the Range of 16 to 25

Valve Type	Valve Size	C_v
Normally–open water valve	1–inch	18
Normally–open water valve	1¼ inch	22.2
Normally–open water valve	1¼ inch	17
Normally–open water valve	1¼ inch	16
Normally–open water valve	1½ inch	25
Normally–open water valve	1½ inch	20
Three–way valve	1–inch	18
Three–way valve	1–inch	18.1
Three–way valve	1¼ inch	16
Three–way valve	1¼ inch	17
Three–way valve	1½ inch	25
Three–way valve	1½ inch	21

At this point, the designer will check the line size of the piping circuit and check the piping size of the reducing fittings. A good choice for the designer would be to show the C_v at 16 and the maximum close-off pressure at 20 psig in this example because there are more control valves available in the range of 16 to 20 (a plus 25 percent deviation) than there are in the rage of 20 to 25 (a plus 25 percent deviation).

d. The C_v calculations for selecting a butterfly valve are identical to those for any valve except that the valve C_v is selected using the calculated C_v at a maximum of 70 degrees open when the valve is to be used in a modulating-control application. (The butterfly valve does not have a good modulating control characteristic curve between 70 degrees and 90 degrees open.)

e. When selecting control valves for liquid service other than water, the designer will take into account the specific gravity of the liquid. If the liquid has a specific gravity of 1.05, for the same gpm, the liquid flow will produce a 5 percent greater pressure drop through the valve.

2-14. Calculation of steam-control-valve flow coefficient (C_v)

a. Calculating and selecting the required C_v of a steam valve requires the designer to consider:
(1) The saturated or superheated condition of the steam.
(2) The inlet pressure at the valve.
(3) The minimum required steam pressure entering the steam-condensing apparatus at peak heating load.

b. the designer will use equations that correct the liquid C_v for the compressibility of the steam. The designer will take into account in his calculations that the limiting factor of steam flow through a control valve is the flow that occurs at critical pressure drop, and will use an appropriate factor for calculating the critical pressure drop. The designer will calculate the critical pressure drop across the valve by using equation 2-5.

$$\text{Pcr} = \text{Km (Pe)} \qquad \text{(eq. 2-5)}$$

Where:
Pcr = Critical pressure drop (psid).
Km = Valve pressure-recovery coefficient (use 0.45)
Pe = Absolute pressure of the steam entering the valve (psia).

c. For steam-control-valve application using saturated steam at a pressure drop across the valve less than the critical pressure drop, the C_v will be calculated using equation 2-6.

$$C_v = \frac{Q}{2.11[(P_e + P_o) \, P]^{1\!/_2}} \qquad \text{(eq. 2-6)}$$

Where:
C_v = Valve flow coefficient (gpm per psid).
Q = Pounds of steam per hour.
2.11 = Compressibility and conversion factor.
Pe = Absolute steam pressure entering the valve (psia).
Po = Albsolute pressure entering steam coil (psia).
P = Pressure drop through valve (psid).

d. When the drop across the valve will be at or greater than the critial pressure drop and the steam is saturated, the C_v will be calculated using equation 2-7.

$$C_v = \frac{Q}{1.74 \, P_e} \qquad \text{(eq. 2-7)}$$

Where:
C_v = Valve flow coefficient (gpm per psid).
Q = Pounds of steam per hour.
1.74 = Compressibility and conversion factor.
Pe = Absolute pressure of entering steam (psia).

e. For superheated steam applications where the drop across the valve is less than the critical pressure, the C_v will be calculated using equation 2-8.

$$C_v = \frac{Q (1 + 0.0007 \, T_{sh})}{2.11 \, [P + P_o) \, P]^{1\!/_2}} \qquad \text{(eq. 2-8)}$$

Where:
C_v = Valve flow coefficient (gpm per psid).
Q = Pounds of steam per hour.
(1+ 0.0007 Tsh) = Superheat factor.
Tsh = Superheat temperature in degress F.

2.11 = Compressibility and conversion factor.
Pe = Absolute pressure of the steam entering the valve (psia).
Po = Absolute pressure of the steam entering the coil (psia).
P = Pressure drop through valve (psid).

Tsh is the Fahrenheit superheat temperature of the steam reduced to the lower pressure from the higher pressure.

f. For superheated steam applications where the pressure drop across the valve at or greater than the valve's critical pressure, the C_v will be calculated using equation 2-9.

$$C_v = \frac{Q (1 + 0.0007 \, T_{sh})}{1.74 \, P_e} \qquad \text{(eq. 2-9)}$$

Where:
C_v = Valve flow coefficient (gpm per psid).
Q = Pounds of steam per hour.
(1+ 0.0007 Tsh) = Superheat factor.
Tsh = Superheat temperature in degrees F.
1.741 = Compressibility and conversion factor.
Pe = Absolute pressure of the steam entering the valve.

g. An example of the calculation required for the C_v of a valve designed to handle saturated steam when the pressure drop will be less than critical is as follows: An air coil requires 250 pounds of steam per hour entering the coil at 2 psig. The steam is generated by a local boiler with the boiler's operating pressure switches set at 5 psig "on" and 8 psig "off". Assuming 5 psig at peak load at the valve inlet and 2 psig at the valve outlet, the C_v is 11.34 as a result of using equation 2-6. In this example, critical pressure drop is not a limiting factor, because the coil pressure at critical pressure drop would be (14.7 + 5) x 0.45 = 8.87 psia, which is not possible unless the system is designed to operate at a high vacuum. A check of manufacturers' literature shows C_v of 10, 12, and 13 and a 1-inch valve. A valve C_v of 12 would be a good design choice for this application.

h. When the calculation for steam-valve selection results in a C_v of greater than 55, the designer will select two valves, with C_v chosen to split the flow unevenly between the valves; a good ratio is one-third of the flow for the smaller valve and two-thirds of the flow for the larger valve, which, however, is not always achievable from the normally available stock of valves. As an example of the calculation required, a steam converter requires 6,000 pounds per hour of steam at 1 psig entering the converter shell at peak load with the steam pressure reduced from 125 psig to 25 psig near the convertor. The critrical pressure is 17.87 as a result of using equation 2-5, and the pressure drop across the valve is 25-1 = 24 psi, which is greater than the critical pressure. The constant-enthalpy process of reducing the steam pressure from 125 psig to 25 psig results in 48 degrees of superheat. The required C_v is

89.1 as a result of using equation 2-9. This C_v is 3 percent larger than it would have been if the steam were saturated at 25 psig and C_v were calculated from equation 2.7. In this example, the designer should show two valves in parallel and attempt to find available valves with C_v of approximately 60 and 30. A check of manufacturers' catalogs for this example results in the designer showing valves with C_v of 70 and 23 and a close-off pressure of 25 psig. The selection results from the following available control valve combinations, any of which will meet the design intent:
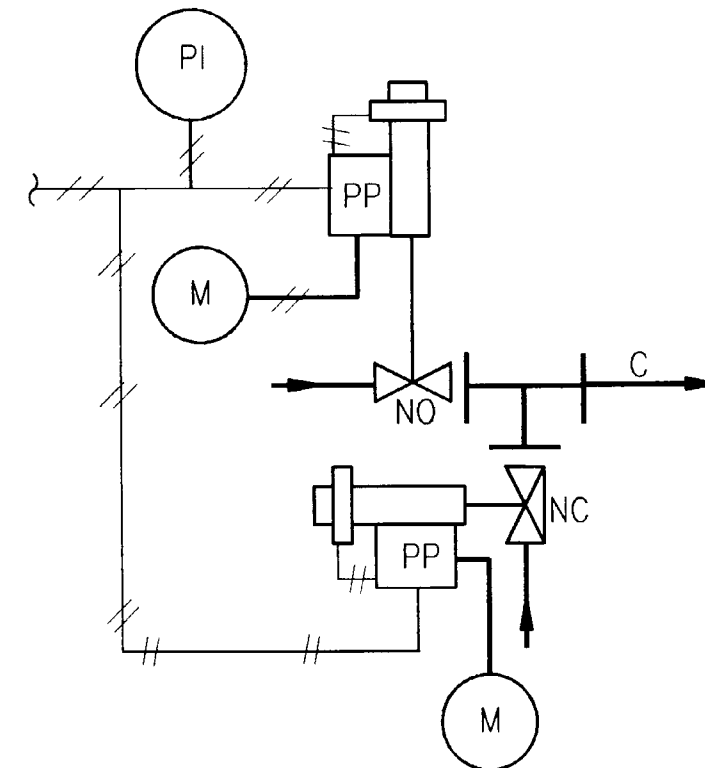
    (1) 1 valve 2½ inch size, C_v = 74' and 1 valve 1½ inch size, C_v = 18, or total C_v = 92

    (2) 1 valve 2½ inch size, C_v = 70, and 1 valve 1½ inch size, C_v = 23, or total C_v = 93

    (3) 1 valve 3-inch size, C_v = 70.7, and 1 valve 1½ inch, C_v = 23.1 or total C_v = 93.8

2-15. Surge-protection provisions for transmitter and control wiring

Because HVAC control-system transmitter signals and single-loop controller signals are interfaced with EMCS, the appropriate surge protection will be provided in the HVAC control system design as described in TM-5-815-2.
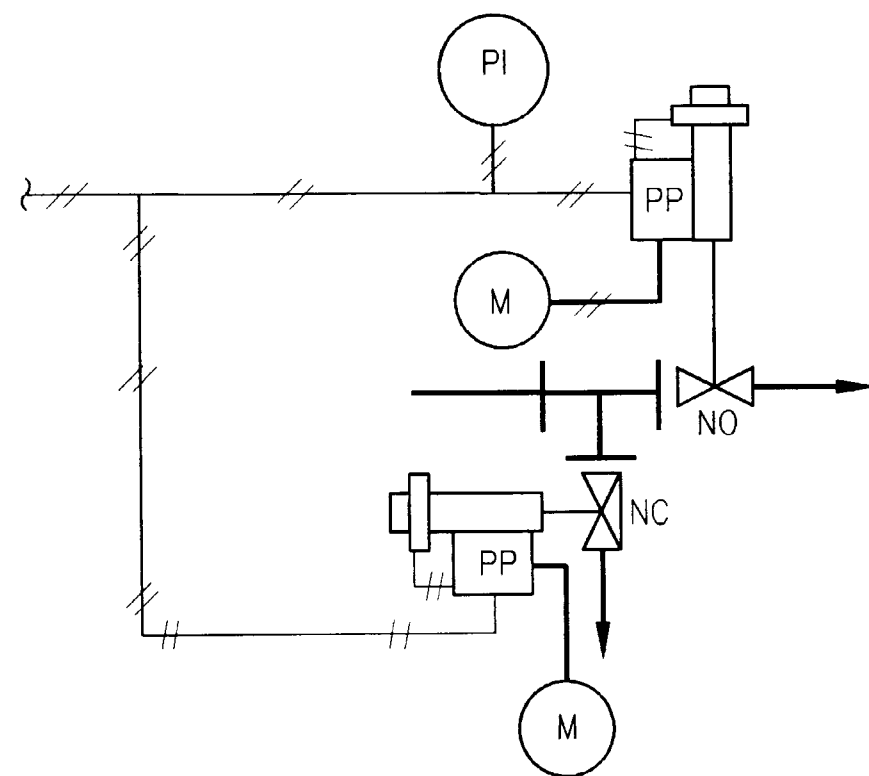
3-WAY BYPASS APPLICATION   3-WAY MIXING APPLICATION

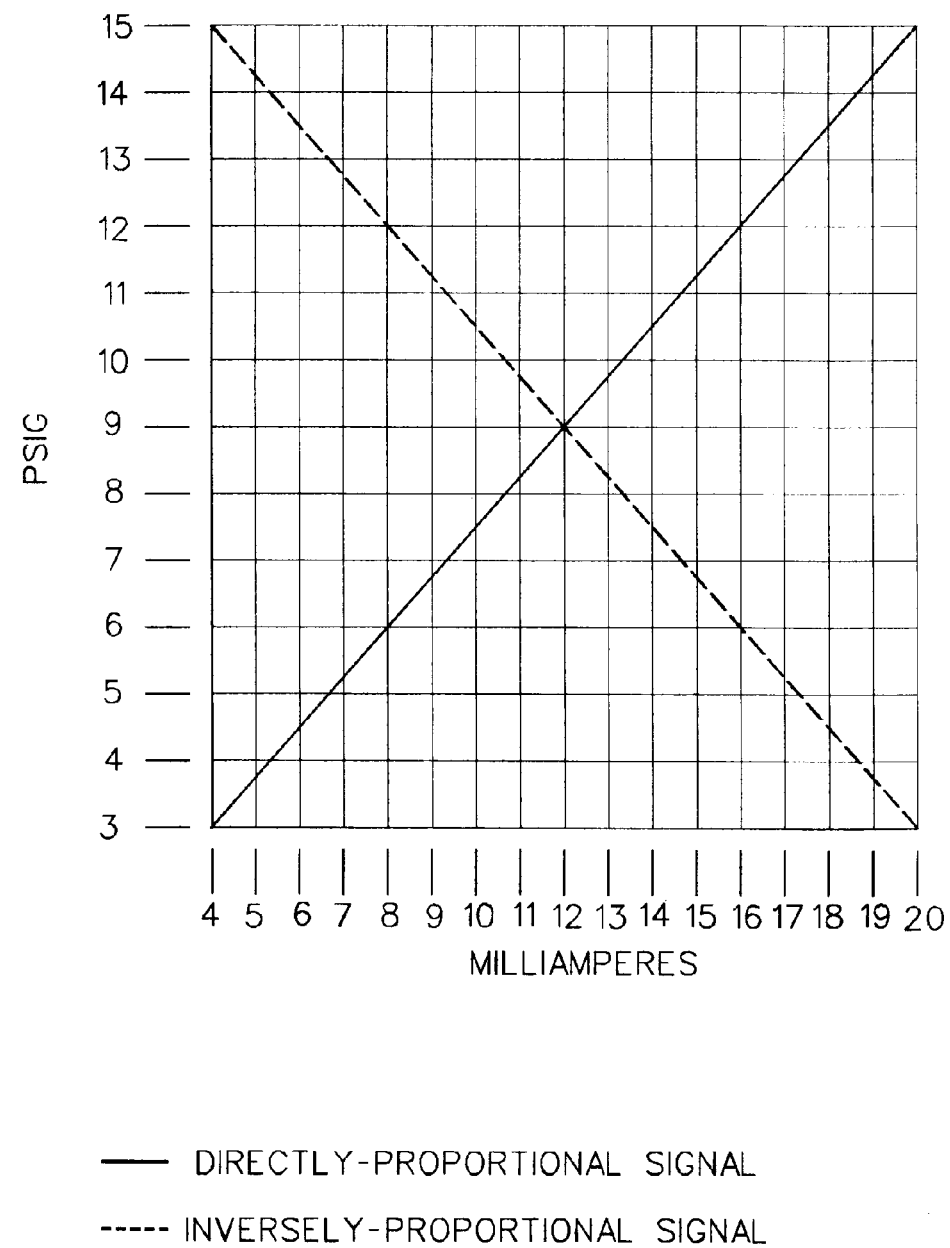*Figure 2-1.   Two butterfly valves on a common pipe tee.*

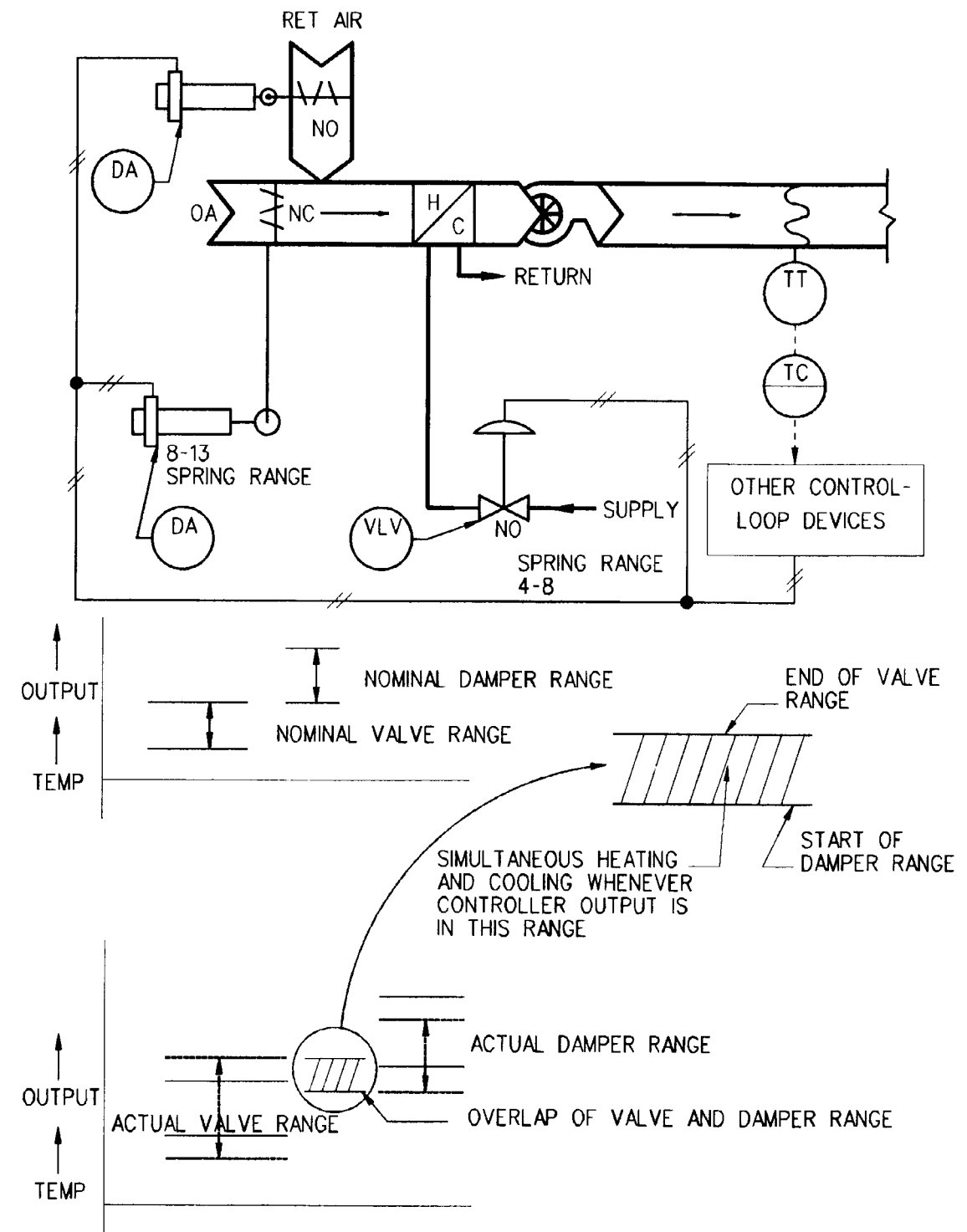Figure 2-2. *Conversion of an electronic signal to a pneumatic signal.*

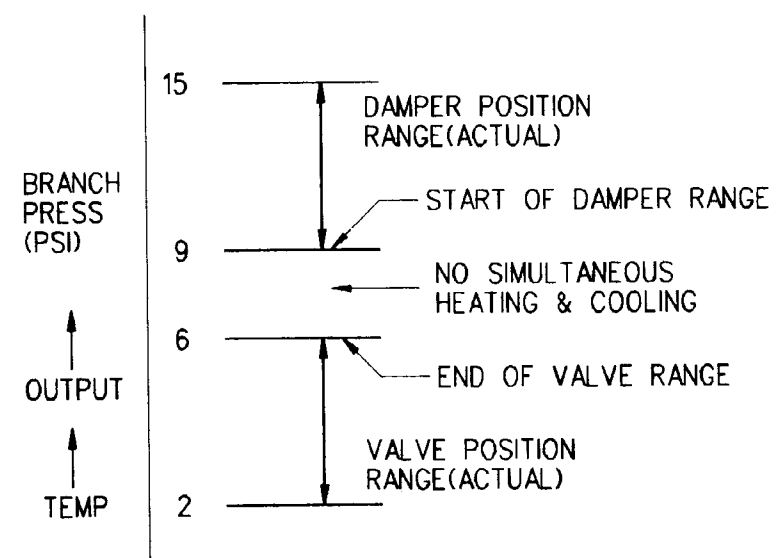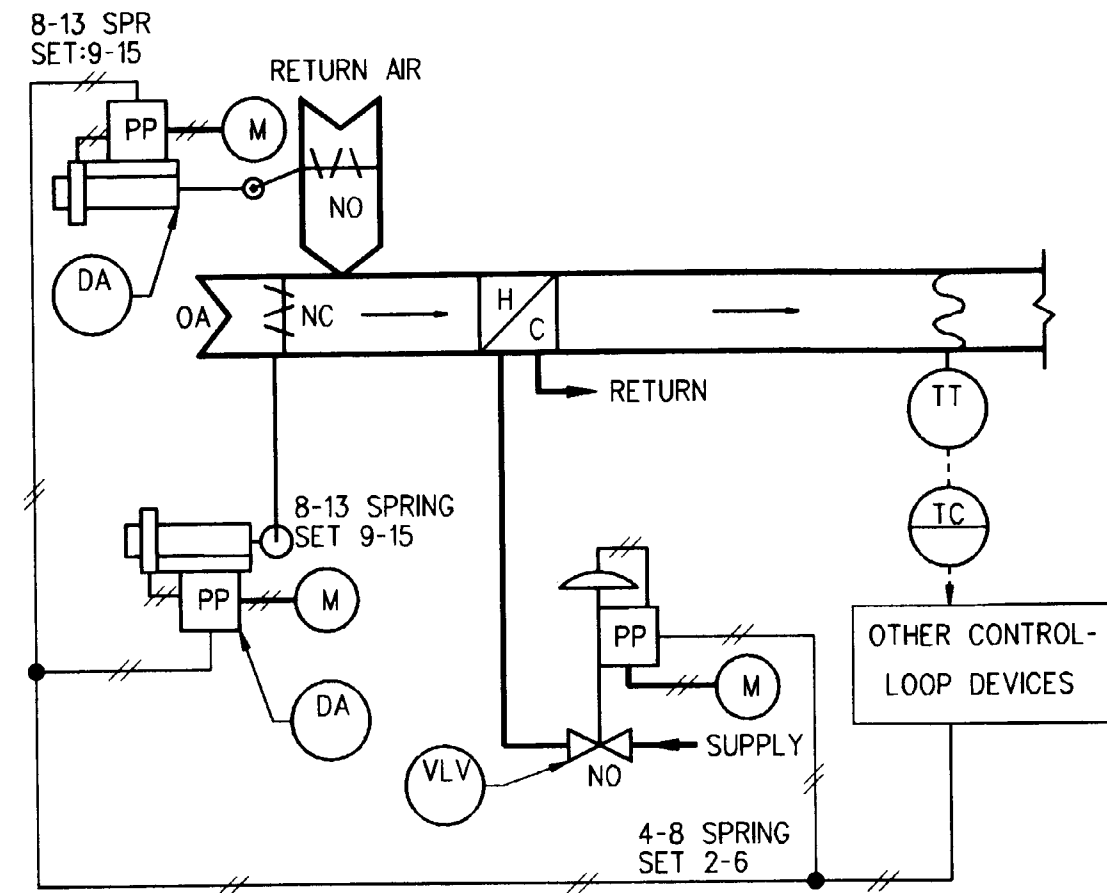Figure 2-3. *Simultaneous heating and cooling with pneumatic actuators without positive positioners.*

Figure 2-4. Control system with positive positioners to avoid simultaneous heating and cooling.
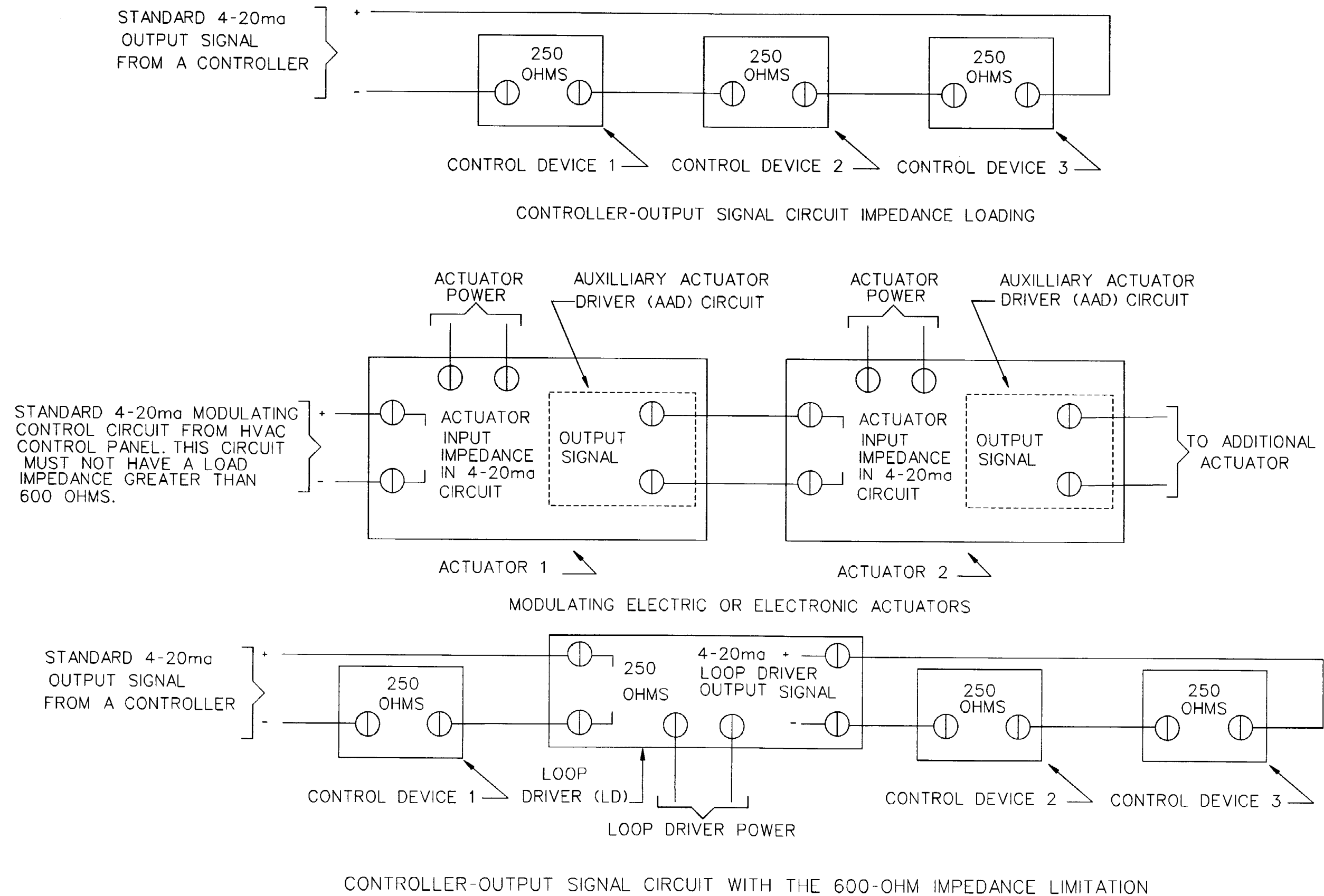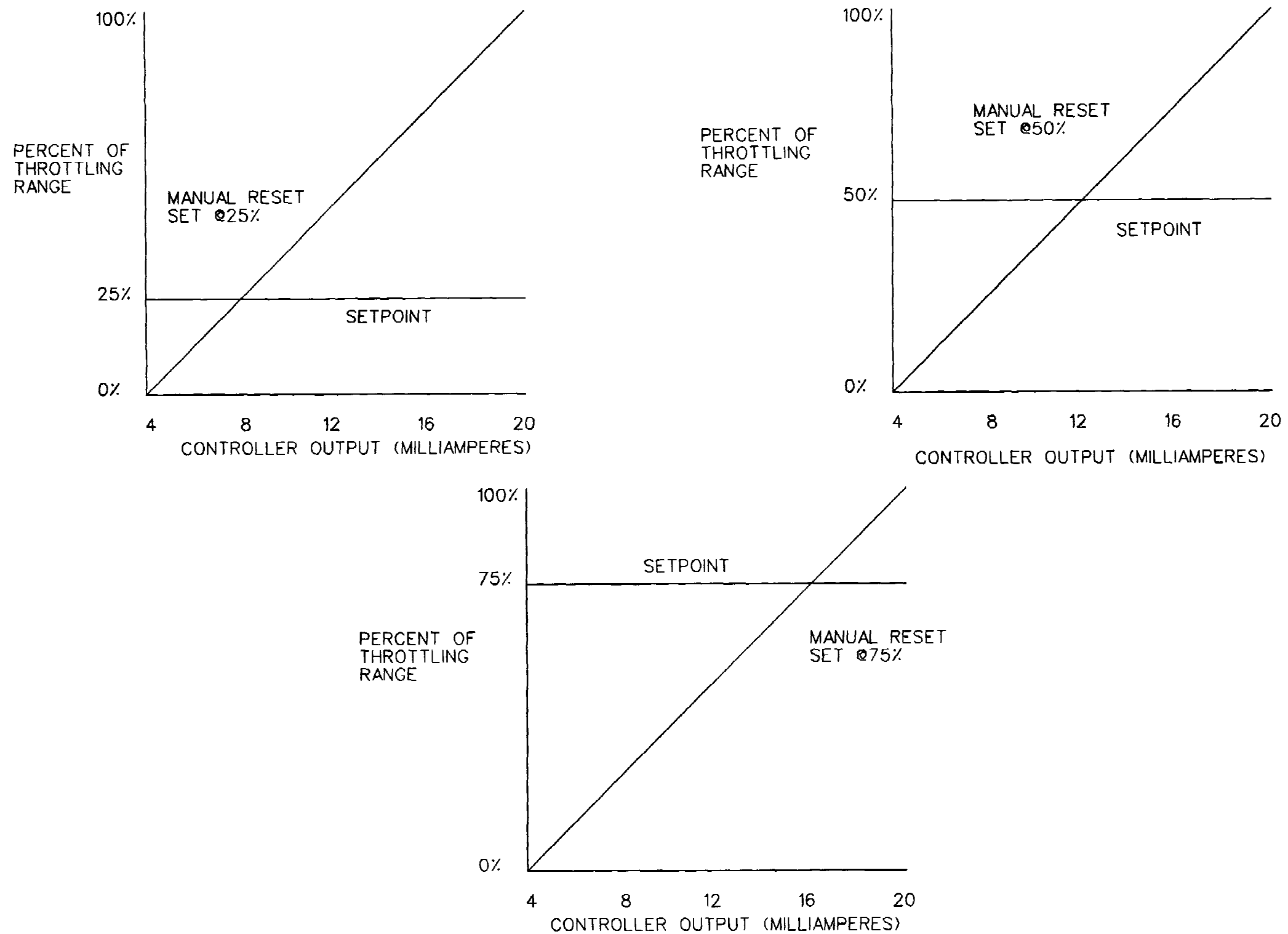
STANDARD 4-20ma
OUTPUT SIGNAL
FROM A CONTROLLER

+

250
OHMS

250
OHMS

250
OHMS

−

CONTROL DEVICE 1 → CONTROL DEVICE 2 → CONTROL DEVICE 3 →

CONTROLLER-OUTPUT SIGNAL CIRCUIT IMPEDANCE LOADING

ACTUATOR
POWER

AUXILLIARY ACTUATOR
─DRIVER (AAD) CIRCUIT

ACTUATOR
POWER

AUXILLIARY ACTUATOR
─DRIVER (AAD) CIRCUIT

STANDARD 4-20ma MODULATING
CONTROL CIRCUIT FROM HVAC
CONTROL PANEL. THIS CIRCUIT
MUST NOT HAVE A LOAD
IMPEDANCE GREATER THAN
600 OHMS.

+

ACTUATOR
INPUT
IMPEDANCE
IN 4-20ma
CIRCUIT

OUTPUT
SIGNAL

ACTUATOR
INPUT
IMPEDANCE
IN 4-20ma
CIRCUIT

OUTPUT
SIGNAL

TO ADDITIONAL
ACTUATOR

−

ACTUATOR 1 → ACTUATOR 2 →

MODULATING ELECTRIC OR ELECTRONIC ACTUATORS

STANDARD 4-20ma
OUTPUT SIGNAL
FROM A CONTROLLER

+

250
OHMS

250
OHMS

4-20ma +
LOOP DRIVER
OUTPUT SIGNAL

−

250
OHMS

250
OHMS

−

CONTROL DEVICE 1 →
LOOP
DRIVER (LD)

LOOP DRIVER POWER

CONTROL DEVICE 2 → CONTROL DEVICE 3 →

CONTROLLER-OUTPUT SIGNAL CIRCUIT WITH THE 600-OHM IMPEDANCE LIMITATION

Figure 2-5. Modulating-control-circuits impedance limitation.

16

Graph 1 (top left):

100%

PERCENT OF
THROTTLING
RANGE

MANUAL RESET
SET @25%

25% — SETPOINT

0%

4   8   12   16   20
CONTROLLER OUTPUT (MILLIAMPERES)

Graph 2 (top right):

100%

MANUAL RESET
SET @50%

PERCENT OF
THROTTLING
RANGE

50% — SETPOINT

0%

4   8   12   16   20
CONTROLLER OUTPUT (MILLIAMPERES)

Graph 3 (bottom):

100%

75% — SETPOINT

PERCENT OF
THROTTLING
RANGE

MANUAL RESET
SET @75%

0%

4   8   12   16   20
CONTROLLER OUTPUT (MILLIAMPERES)

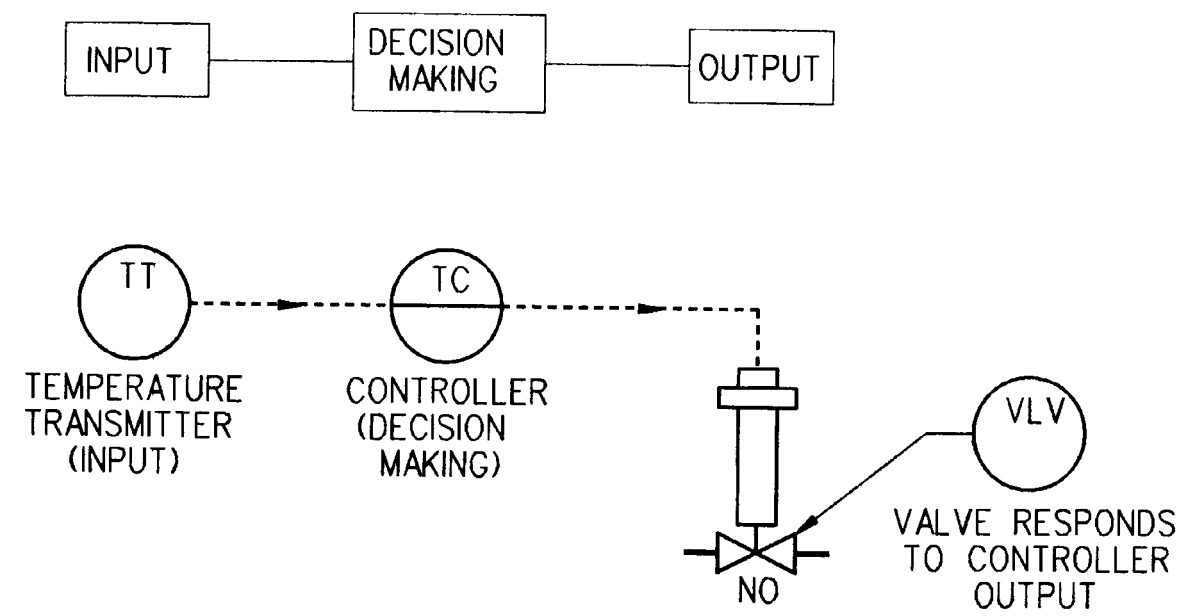MANUAL RESET FEATURE

Figure 2-6. Manual reset feature.
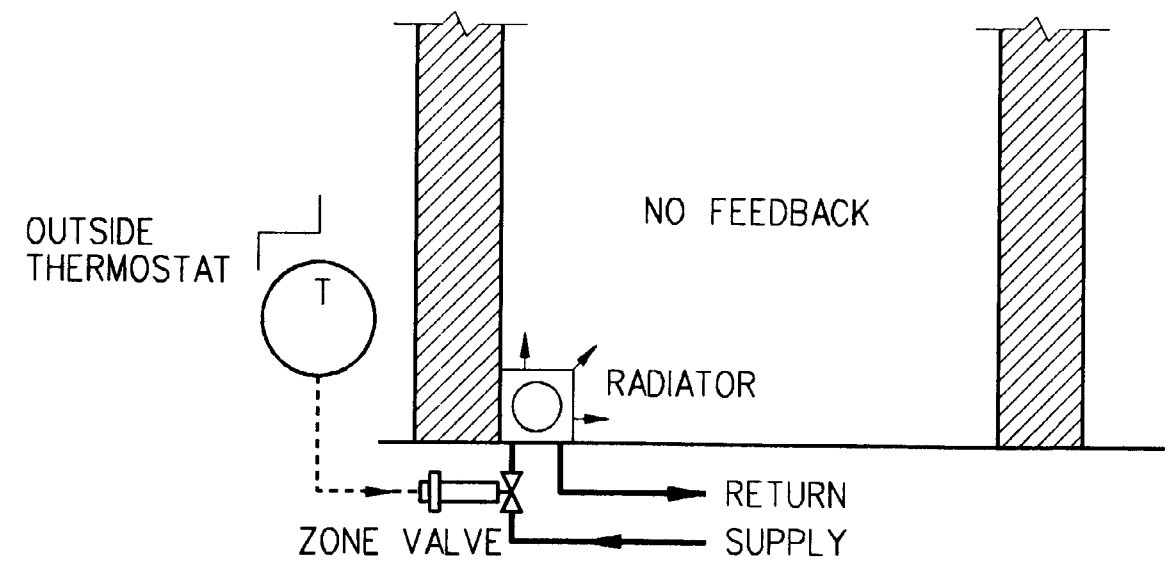
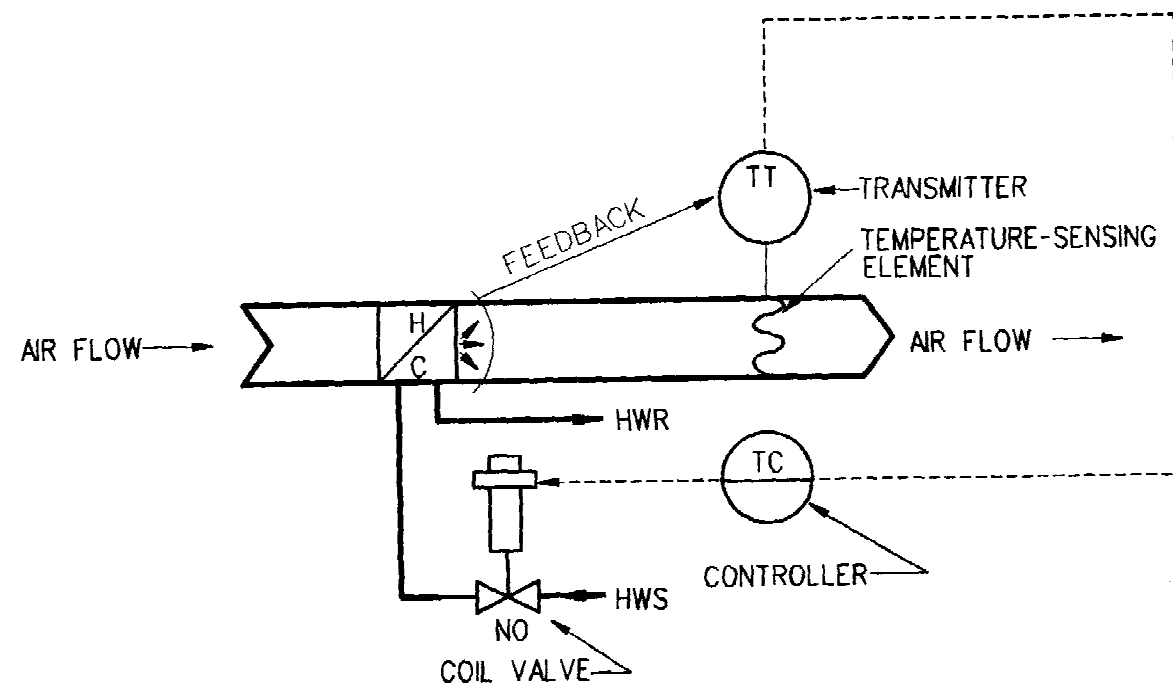Figure 2-7. Control loop.

Figure 2-8. Open control loop.
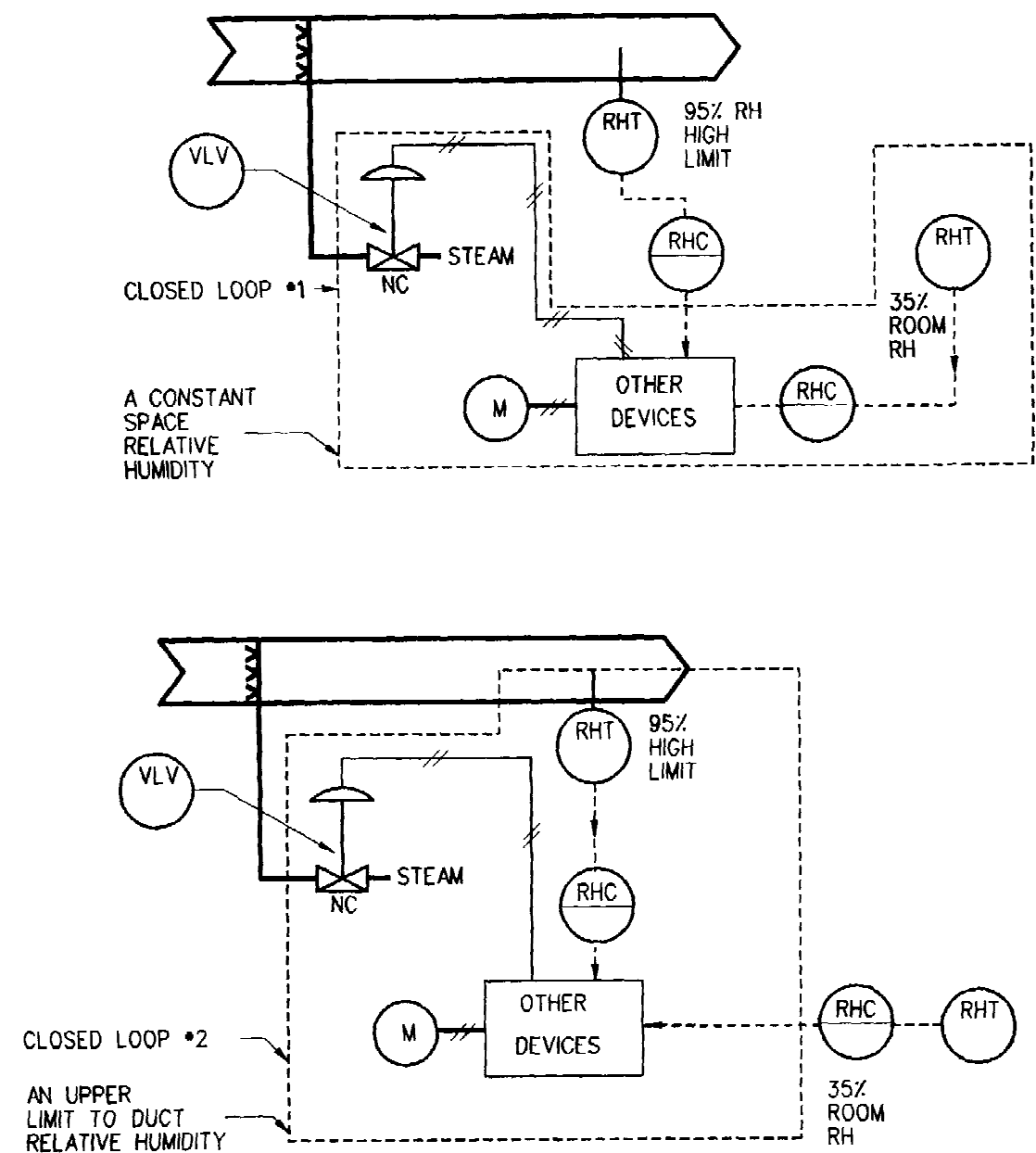
Figure 2-9. Closed control loop.

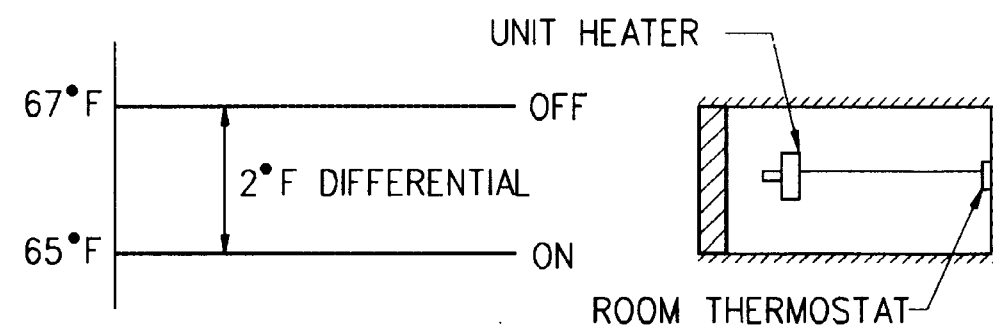Figure 2-10. Two loops controlling one device (humidity control with high limit).
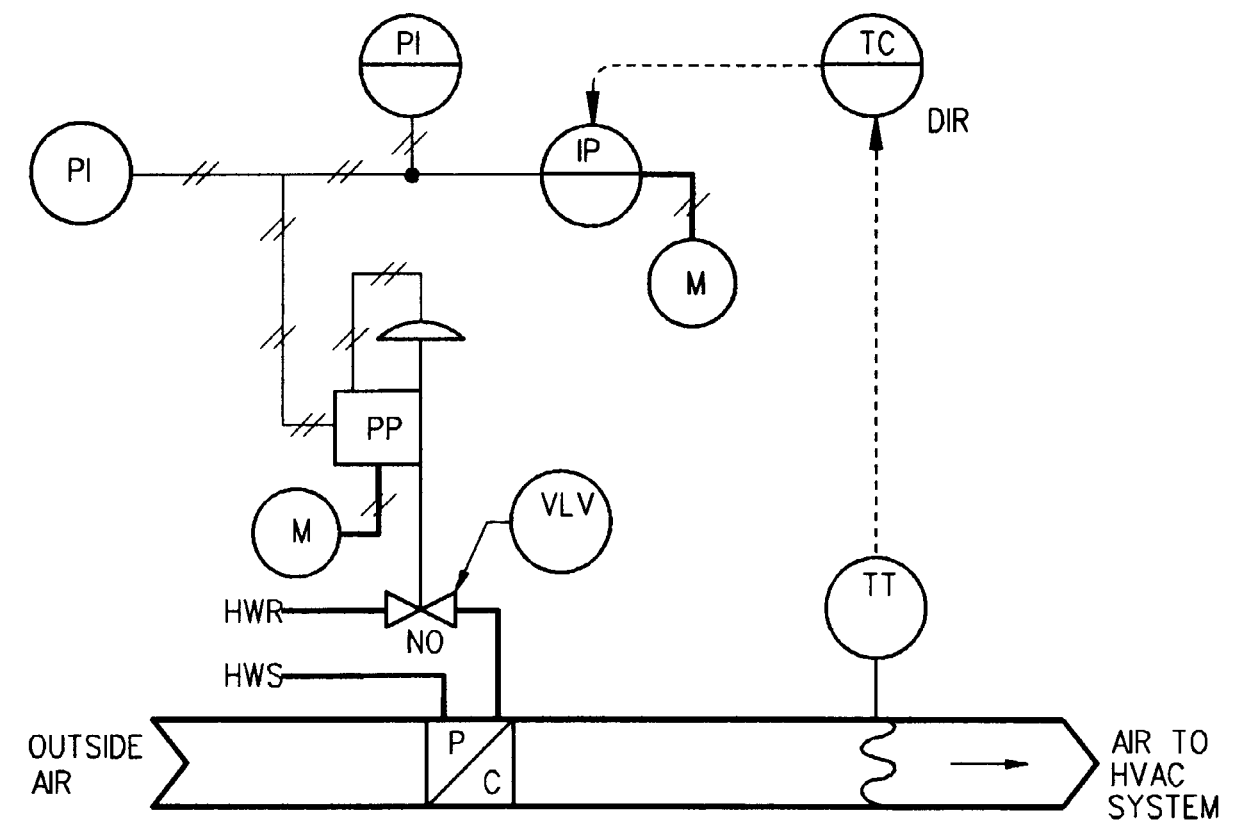
Figure 2-11. *Two-position control.*

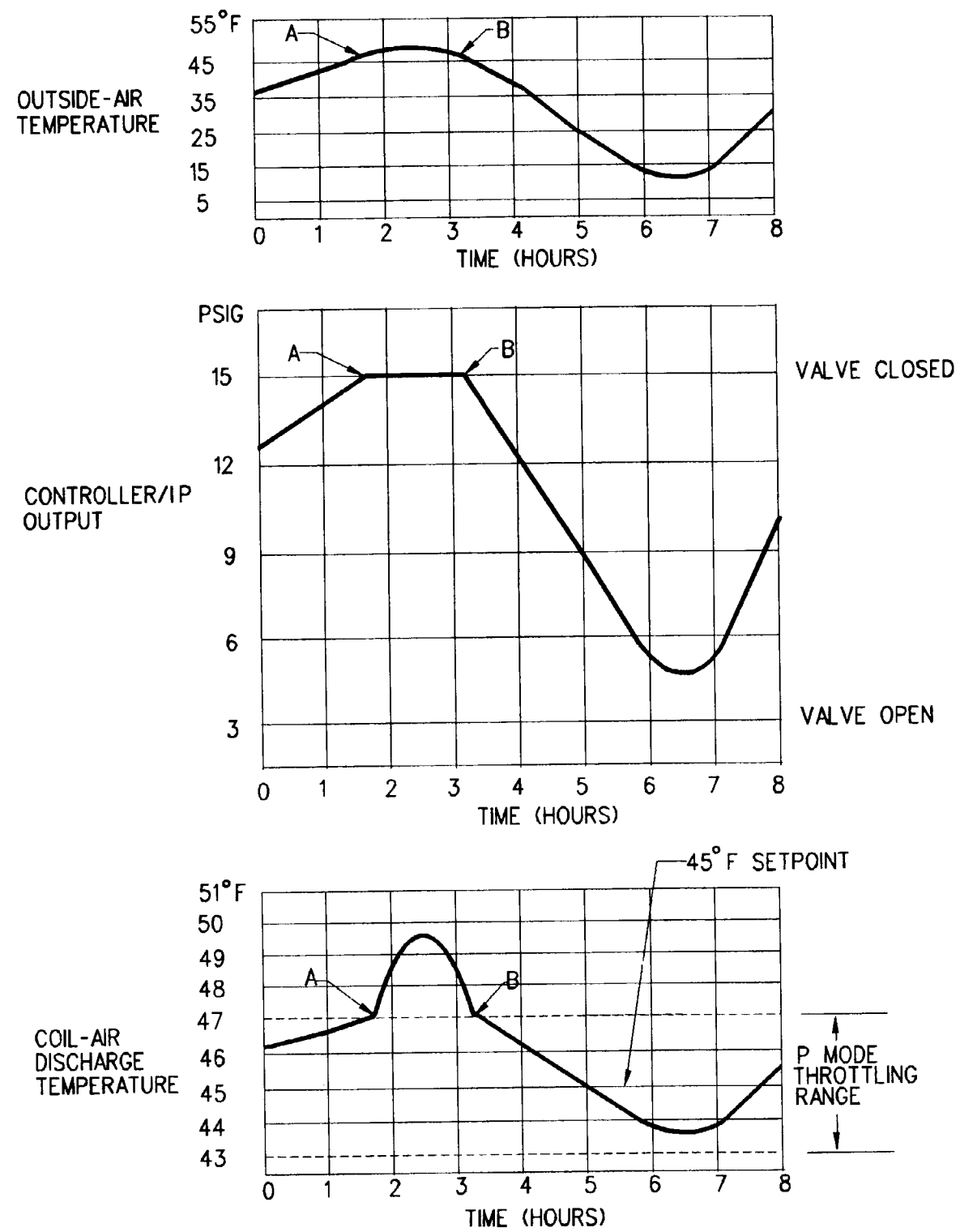Figure 2-12. *Simple control loop applied to outside-air heating.*
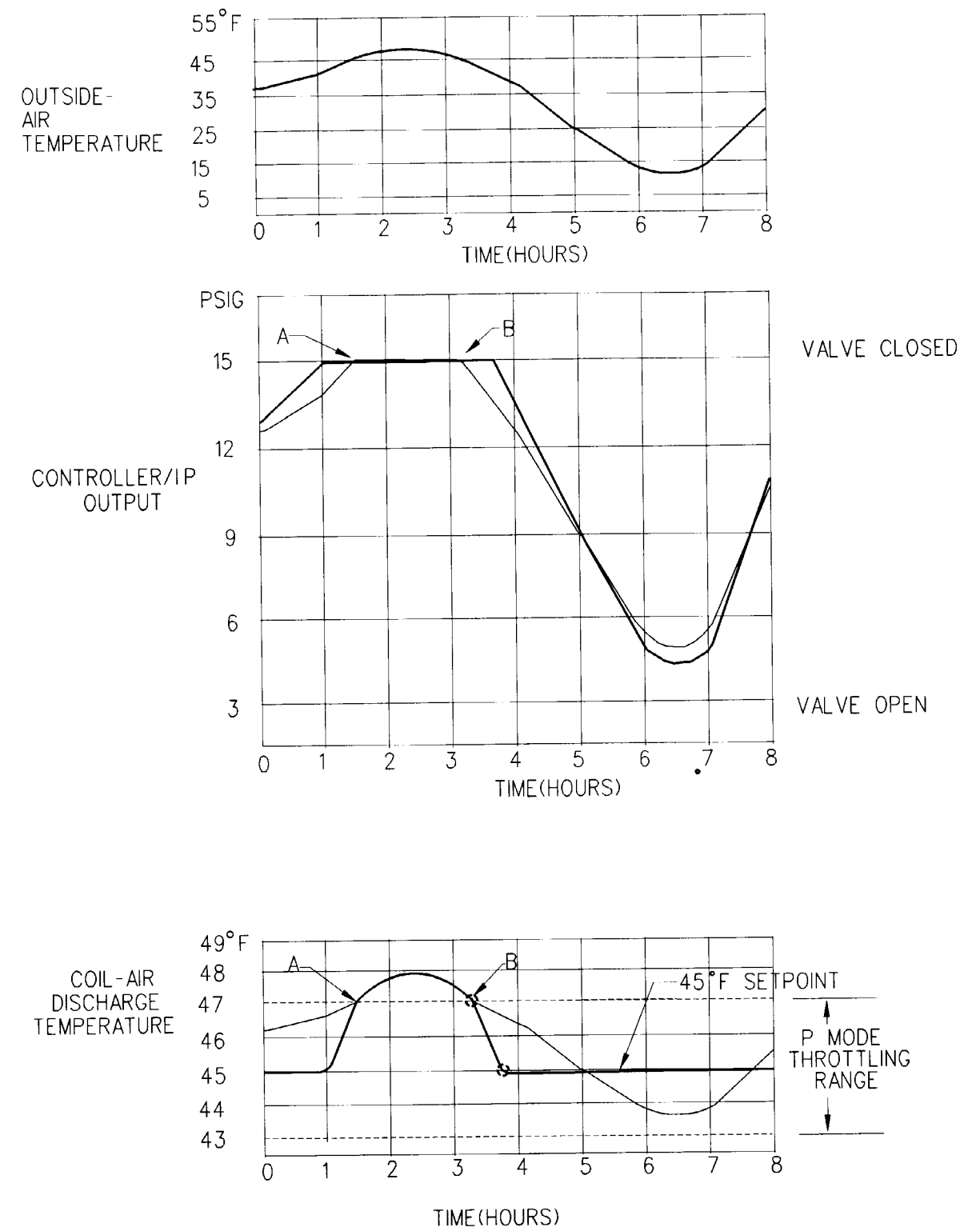
Figure 2-13. Proportional-control mode.

Figure 2-14. Proportional-plus-integral control mode.

PROPORTIONAL CONTROLLER

SETPOINT
STEP CHANGE IN SETPOINT
P OUTPUT IS PROPORTIONAL TO PROCESS VARIABLE
OFFSET
NEW SETPOINT
TIME TO STABILIZE

COMPARISON OF CONTROL OUTPUTS FOR THE SAME STEP CHANGE

SETPOINT
P OUTPUT
STEP CHANGE IN SETPOINT
NEW SETPOINT

PROPORTIONAL+INTEGRAL CONTROLLER

SETPOINT
STEP CHANGE IN SETPOINT
PI OUTPUT IS PROPORTIONAL TO PROCESS VARIABLE AND TO OFFSET (UNTIL OFFSET IS ZERO)
NO OFFSET
NEW SETPOINT
TIME TO STABILIZE

SETPOINT
P OUTPUT
I COMPONENT ADDITION
PI OUTPUT
STEP CHANGE IN SETPOINT
NEW SETPOINT

PROPORTIONAL+INTEGRAL+DERIVATIVE CONTROLLER

SETPOINT
STEP CHANGE IN SETPOINT
PID OUTPUT IS PROPORTIONAL TO PROCESS VARIABLE, OFFSET, AND RATE OF CHANGE (UNTIL OFFSET AND RATE OF CHANGE ARE ZERO)
NEW SETPOINT
TIME TO STABILIZE

SETPOINT
PI OUTPUT
D COMPONENT ADDITION
PID OUTPUT
STEP CHANGE IN SETPOINT
NEW SETPOINT

COMPARISON OF P MODE, PI MODES AND PID MODES FOR A STEP CHANGE IN SETPOINT AND THE CONTRIBUTIONS OF EACH MODE TO CONTROLLER OUTPUT SIGNAL.
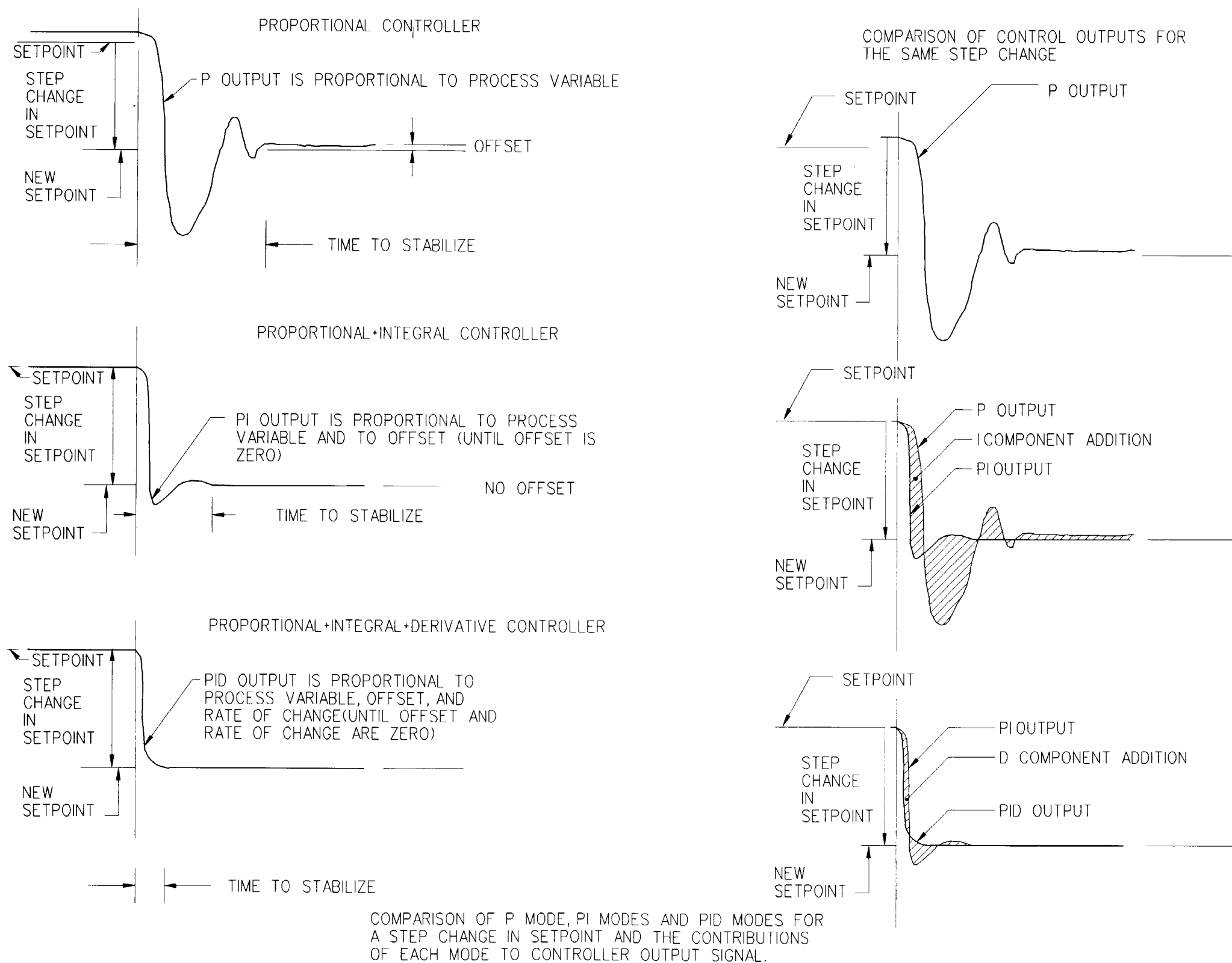
Figure 2-15. Comparison of P mode, PI modes and PID modes for a step change in setpoint, and the contributions of each mode to controller-output signal.